DESIGN OF A TEST INSTALLATION FOR INVESTIGATION OF REYNOLDS' NUMBER EFFECT ON GAS TURBINE PERFORMANCE

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REYNOLDS' NUMBER EFFECT ON GAS TURBINE PERFORMANCE,

DESIGN OF A TEST INSTALLATION FOR INVESTIGATION OF

by

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Professor Joseph S. Newell
Secretary of the Faculty
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Cambridge, Massachusetts

Dear Professor Newell:

We take pleasure in submitting herewith a thesis entitled "Reynolds' Number Effect on Gas Turbine Performance, Design of a Test Installation for Investigation of", in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

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The authors have been assisted considerably in the preparation of this thesis by the cooperation of various persons on the staff of the Gas Turbine Laboratory, the Mechanical Engineering Department and the Aeronautical Engineering Department. Professor E. S. Taylor has contributed unstintingly with his advice and experience, without which the authors would have been sorely handicapped. The requested advice of and analysis by Professor A. R. Rogowski relative to the feasibility of applying an Eddy Current Brake as a dynamometer was greatly appreciated. Mr. F. Lustwerk of the Gas Turbine Laboratory provided valuable information relative to his experience in use and operation of the wind tunnel. Mr. A. H. Redding of the Westinghouse Electric Corporation contributed an afternoon of his time, and provided a basis for the final choice of dynamometer type. He was instrumental in providing drawings and specifications of a similar device owned by the company. To such individuals, mentioned or otherwise, the authors wish to extend their deeply-felt gratitude.

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1.0 INTRODUCTION

1.1 It is of vital concern to aviation interests to be able to make predictions of gas turbine performance with due regard to the effects of geometric scale and/or changes in altitude, which are directly related to Reynolds' Number. Due to the large quantities of air involved in the gas turbine cycle in actual practice, it is extremely expensive to construct wind tunnels, or other such test facilities, for the calibration or testing of such power units at varying Reynolds' Number. As a stop-gap measure, it has been the practice of designers to estimate performance of proposed engines on the basis of geometric and dynamic similarity to smaller or similar existing engines. Altitude performance has been predicted from sea level tests on actual engines in terms of dynamic similarity in speeds, temperature and pressure levels, etc. These predictions generally have disregarded the possibility that component efficiencies may vary for the conditions of such extrapolations. Cascade tests have provided a two-dimensional basis for correcting for induced aerodynamic losses, with fairly well substantiated empirical correction factors to three-dimensional conditions, (ref. (a)). However, airfoil profile losses, which are more directly related to Reynolds' Number, have not been so accurately determinable, especially over the possible range of operation at altitude.

1.2 The Gas Turbine Laboratory at MIT has in its possession a small scale turbo-jet engine manufactured by Westinghouse Electric Corp., namely the Model XJ32-WE-4 (X9.5B). This engine provides a single-stage turbine with a blade-tip diam. of 8.1 in. and a root diam. of 5.1 in. The variable-density wind tunnel at the above activity provides a test facility in which the performance of such a turbine could be studied in detail, over

a range of Reynolds' Numbers from 253,500 to 25,350 based on blade chord and turbine inlet conditions. (see Appendix D). This would extend the range of investigation reported in ref. (a) by approximately 25 per cent on the upper limit and 33 per cent on the lower limit of that of ref. (a). It is the purpose of this project to provide a design for the required test installation and associated equipment so that the above facilities may be used in the investigation of the effect of Reynolds' Number variation on turbine performance.

2.0 SUMMARY

2.1 Figures 3 to 26 show the general details of the resulting design. The necessary assembly and detail drawings for existing Westinghouse parts are available in the Gas Turbine Library and are described in ref. (h). It will be seen that provisions have been made for a horizontal installation in the low speed branch of the variable density wind tunnel on the second floor of the Gas Turbine Laboratory, MIT. The turbine wheel has been located as close as possible to an elbow adapter in the tunnel so that the power take-off could be taken through the elbow. This results in a condition whereby the air stream is delivered to the turbine very shortly after the turn through the elbow, but any other installation which would provide an external location for the dynamometer would necessitate excessively long extension shafts, which would be unacceptable from dynamic considerations. However, due to the rather severe acceleration and diffusion in the airstream through the compressor diffusor which was used to facilitate mounting, it is believed that flow at entrance to the turbine nozzles will be sufficiently uniform, in both

direction and degree of turbulence, as to be representative of actual "hot" running conditions at the turbine.

2.2 The installation shown will permit safe operation of the turbine at speeds from zero to somewhat in excess of 18200 rpm. (see Appendix A for shaft extension dynamics). The latter speed corresponds by dynamic similarity to an actual ("hot") running speed of 34000 rpm. (see Appendix B). Maximum desired running speed is 18,220 rpm. Static inlet pressure as delivered by the tunnel is controllable from 22.4 psia to 2.24 psia with mass flow of 7.005 to .7005 lbs./sec., permitting power variation in the ratio 10 to 1 at any turbine speed. Total temperature at tunnel compressor outlet is controllable from 80° to 100° F., corresponding to a static temperature range of from 73° to 93° F. at the turbine inlet. The possible range of investigation will be from a Reynolds' Number of 25,350 to 253,500.

2.3 Assembly directions for the test installation will be found in Sections 5.0 to 5.26 inclusive. Detailed operating instructions have been based on the various considerations of the proposed design, and on safety. They will be found in Sections 6.0 to 6.34 inclusive. It is to be noted especially that operation at speeds over 18200 rpm. will most probably be safe, since that represents the critical speed for the new turbine shaft extension, assuming zero fixity at the ends. The assumption is considered to be very conservative. All appendices have been prepared for a top speed of 20,000 rpm., at which it is recommended that the safety governor mentioned in Section 3.5 be set, thereby providing a margin of 1,780 rpm. over maximum desired operating speed, for adjustment purposes during running.

3.0 DISCUSSION

3.1 In providing a facility for measurement of performance of the turbine of the WE X9.5B Turbo-Jet engine, the first and most obvious need was for a device capable of absorbing and measuring an output of 250 hp. at 20,000 rpm. (see Appendix F), with possible variation in output in the ratio of 10/1 at any given speed. Such devices are not presently available at MIT, nor are they procurable commercially except, probably, on a developmental basis. Therefore it became necessary to design such a dynamometer locally. Several basic techniques were considered, all of which would lend themselves to direct coupling with the turbine (a very desirable and necessary feature), and are presented in the following paragraphs:

3.11 A plain disc running submerged in water, to absorb the required power through skin friction and circulation effects. The variation in power absorption could be controlled by "water level" (amount of water) on the disc, since any water in the disc chamber will be thrown as far out on the disc as possible by centrifugal effects. This technique was discarded when it was shown (Appendix F) that, although the maximum power could easily be absorbed by one submerged disc of small O.D., there would very likely be instability in operation at low power input at high speeds. The location of the inner radius of the water was very critical under such conditions. (Power varies approximately with fifth power of radius, third power of speed).

3.12 The above led to consideration of the possibility of using a simple cylinder rotating about a vertical axis in water of variable

depth. Power absorption would vary linearly with the submerged length of the cylinder (Appendix G). However, Appendix G shows that a shaft of excessive length would be required if any degree of stability were to be achieved at low power settings. Its vibration characteristics would probably be unacceptable, as seen from the analysis for the turbine shaft extension (Appendix A). Furthermore, a pressure-tight enclosing cylinder would be required to prevent the water from being thrown away uncontrollably in excess of that amount required for cooling. This would probably result in a violent circulation in the container, with possible instability, not readily analysed mathematically. For these reasons, the technique was considered to be not usable, and was discarded.

3.13 Power absorption by pumping water through a centrifugal water wheel. Since the water leaving the wheel would be thrown away or used for recirculation, input would probably be measured most easily by some sort of torque-meter on the connecting shaft. Water would be introduced into the wheel axially as near the axis as possible, and its momentum would be changed by producing tangential velocity at the rim. Appendix H shows that for reasonable flows, tangential velocities would be such as to cause unacceptable erosive action on any enclosing casing with which it might come in contact. A visit to the Gas Turbine Division of Westinghouse at their South Philadelphia Works revealed that that company had conducted tests to determine the severity of such erosion. They directed a jet of water normal to a 1/16-in. thick stainless steel plate mounted on a wheel

which produced a relative velocity of 1000 ft./sec. The plate was severed within a period of five seconds. Although it is considered possible that other materials could be found which would show better durability in this desired application, it would be optimistic to believe that one could be found which would have acceptable service life. Therefore this technique was discarded.

3.14 A device which has shown very satisfactory results locally as a power absorption device is the Eddy Current Brake. This device consists of a metal disc rotating in electrically induced magnetic fields. The heat produced in the plate by the eddy current generation is carried away by water jets, usually directed through the cores of the electromagnetic coils. Ref. (b) gives the operating characteristics for such brakes with wheels of magnetic materials. More desirable high-speed characteristics are obtained when non-magnetic materials are used in the wheels. Although there has been no local experience with such brakes at the desired top operating speed, Prof. A. R. Rogowski of the Mechanical Engineering provided an extrapolation of some of his test results on brakes of several steels at speeds up to 10,000 rpm. This extrapolation predicted the following design provisions for the subject problem:

Material	Disc Size (Diam.)	No. Coil Pairs	Amp-turns/coil pair
18-8 Stainless Steel	18"	4	14,000
18-8 Stainless Steel	14"	5	14,000
Steel (Boiler Plate)	14"	8.4	14,000

However, it is shown in Appendix F that the necessary flow of water required for normal cooling in a water brake was 12.71 gal./min.. In the

which produced a relative velocity of 1000 ft./sec. The plate was exposed within a period of five seconds. Although it is doubtful [illegible] that other materials could be found which would give better [illegible] in this desired application, it would be [illegible] to believe that one would be found which would have [illegible] service life. Therefore this technique was discarded.

3.13 A device which has given very satisfactory results [illegible] is a [illegible] device is the High Current Brake. This device consists of a metal disc rotating in electrically induced magnetic fields. The heat produced in the plate by the eddy current generation is carried away by water jets, usually directed through the cores of the electromagnetic coils. Ref. [illegible] gives the operating characteristics for such brakes with discs of magnetic materials. [illegible] high-speed characteristics are obtained when non-magnetic materials are used in the wheels. Although there has been no local experience with such brakes at the desired top operating speed, Prof. [illegible] of the Mechanical Engineering Department [illegible] an extrapolation of some of his test results on brakes of various sizes at speeds up to [illegible] rpm. This extrapolation predicted the following design provisions for the subject problem

Material	Disc Dia. (Inches)	[illegible]	[illegible]
18-8 Stainless Steel	[illegible]	8	[illegible]
[illegible] Steel	[illegible]	[illegible]	[illegible]
Steel [illegible]	[illegible]	8.8	[illegible]

However, it is shown in Appendix [illegible] that the necessary flow of water required for normal cooling in a water brake was [illegible] gal./min.. In the

Eddy Current Brake it is believed that the transmission of heat to the cooling water would be considerably less effective than in the simple water brake, resulting in a much higher flow requirement. This water would be subjected to considerable centrifugal pumping on the brake wheel surface, at radii of not less than 7-in., making for considerable unmeasurable losses. Further, windage losses alone on a wheel of such size would not be acceptable if not measurable. These problems could probably be obviated if power input were to be determined by torque measurements in the shaft, but such measurements are of questionable accuracy. Further, there would be the above-mentioned erosion problem of collecting water moving at high velocities. This technique was shelved therefore, in favor of one which showed more promise, discussed below.

3.15 During the above-mentioned visit to Westinghouse, it was learned that that organization has been using successfully a Froude-type water brake for applications similar to ours. This device is in effect a 100%-slip hydraulic clutch, with power absorption controllable by varying the amount of water in the casing, cooling to be provided by continuous replacement. Fig. 1 provides typical performance characteristics obtained by Westinghouse. The plot was obtained by extrapolation of the test data for much larger but slower similar brakes, adjusting scale by the fifth-power rule for power variation with radius, and the third-power rule for variation with rotational speed. Actual test results agreed very satisfactorily with the predictions on this basis. The characteristics desired in this project could be obtained by similar

extrapolation of the wheel and stator dimensions by the same rules. The company has delivered to M.I.T. the detailed drawings and performance data for this brake. They pointed out that in no way did they guarantee the design, performance, or safety of the device, nor are they on the market to sell such equipment. It is considered that it is within the capabilities of local facilities to manufacture such a device, using the general details provided by Westinghouse. It could be adapted in such a way that a "family" of characteristics similar to that of Fig. 1 could be obtained simply by providing different sets of power wheels and stators, with adapters which would give the proper casing voids. Although the production cost of such a brake is considerably greater than it would have been for some of the techniques described above, this is considered to be justified by the characteristics obtained and their reliability, and the stability and controllability of the device over the desired range.

3.2 In the interests of economy it was decided to use as much of the available turbo-jet engine as possible in the desired installation, to minimize the number of new parts which would have to be made. The construction of the engine permitted a complete separation of components just behind the compressor at the shaft coupling and forward face of the compressor diffusor flange. This left the compressor diffusor, combustion chamber, bearings support assembly, and turbine inlet nozzles available as a complete assembly in which the turbine sub-assembly could be installed complete with two radial roller bearings. The lubrication distribution system for the two bearings is integral with the resulting unit. However, since the thrust bearing for the entire turbo-jet unit

was incorporated in the compressor assembly, it became necessary to provide a replacement when the forward part of the engine was discarded. This has been provided external to the tunnel, and has been incorporated on the coupling of the turbine shaft extension, as shown on Fig. 3, in order to minimize turbine shaft extension length. (See Appendix I for analysis of Thrust Bearing requirements.) A sealed bearing is used to eliminate the lubrication problem and the possibility of contaminating the tunnel through the tunnel seal. The fuel nozzles and combustion chamber liner were removed and replaced with new inner fairings (Figs. 15 & 16) to conduct the airstream to the turbine with as little loss and turbulence as possible.

3.21 Normal lubrication of the two roller bearings (Nos. 2 and 3 of the turbo-jet engine) is by an oil-air mist produced by a pump (Westinghouse Part No. 23F530-2) and Mixer (Westinghouse Part No. 14G210-line 1). working off the accessory section. Since the accessory section has been discarded along with the forward part of the engine, it will be necessary to mount the pump and mixer somewhere external with an auxiliary drive for the pump (an electric motor at 1750 rpm. would be adequate). This should be mounted as close to the test installation as possible. Air for the mixer can be drawn from the room through a reducing valve by which pressure in the bearing support will be controllable. The scavenging system described below will provide the pressure differential required for flow. Ref. (h) shows the necessary piping and connections, which should be duplicated as nearly as possible in the test installation. The oil-air mist line should be introduced through the turbine support flange in the tunnel by means of a fixed gas-tight fitting.

3.22 In normal operation the oil-air mist is discharged from the bearings into the exhaust and thrown away. However, it is not permissable to discharge any contamination into the wind tunnel system because of cleanliness requirements when Schlieren apparatus or Interferometers are to be used. Therefore, some sort of an auxilary scavenging system had to be devised. It was seen that the turbine bearings and assembly could be isolated within the compressor diffusor and turbine bearing support assembly. A labyrinth seal for the turbine shaft and a gasket is mounted on the after end of the bearing housing in place of the old radiation shields (Westinghouse Part Nos. 14G963-1&2), which were discarded. The forward end is sealed by means of a new nose cone, (Fig. 21), which seals on the diffusor casting in place of the former diffusor cone and seal (Westinghouse Part No. 19E269 line 1&2), and on the new shaft extension by a labyrinth seal incorporated in the new nose cone. Sharp crests are provided in the seals to provide for "running in" to minimum clearance on the shaft. The rubber gasket at the after end is needed to cover existing screw holes and other cutaways in the after bearings housing flange, and to effect the seal between the bearings housing and the bearings support. It is to be made oversize relative to the I.D. of the support tube, so that the necessary deformation during assembly will effect the seal. An inward flow must be established at all seals and joints, to ensure that no oil gets into the tunnel. Scavenging lines cut into the bearings support (see Fig. 17) will satisfy this requirement when connected to a large-capacity ejector available in the laboratory. The combustion chamber must be turned from its normal installed position relative to the diffusor so

that one of the spark plug holes which provide access for the evacuation lines will be located as nearly at the bottom as possible. See paragraph 5.1 (c). The lines have been made as big as is consistent with required passage through the existing spark plug holes in the combustion chamber casing, to provide as much insurance as possible that available flow will exceed any leakage through the various joints and seals.

3.3 One of the major decisions as to how the test installation could best be made concerned the manner and location of the turbine in the wind tunnel system. It would have been the "cleanest" installation if the turbine and dynamometer could have been mounted as a unit entirely within the tunnel. This would have eliminated the need for a turbine shaft extension, since the dynamometer could have been coupled directly to the existing turbine shaft coupling. However, it is obvious that a dynamometer operating at the required speeds must depend on some sort of a fluid medium. The requirement of no contamination in the tunnel therefore ruled out this possibility, since the problem of perfectly sealing the dynamometer would surely have been a major stumbling block. The logical alternative was to locate at one of the elbows in the tunnel, so that the dynamometer, externally mounted, could be coupled to the turbine shaft extension projecting through the wall of the elbow. Rather than cut up the existing tunnel elbow at the desired location, it was decided to completely replace it with a new design. This gives the added advantage of designing for shortest possible shaft extension. The outermost passage diameter at inlet to the diffuser fortunately allowed direct mating to a 8-in. nominal-size pipe, and so, since the entire regular elbow required replacement, 8-inch pipe and fittings are used wherever

possible. Three elbows or types of turn were available, a standard 8-inch "short-radius" elbow, with an eight-inch turning radius on the centerline; a standard 8-in "long radius" elbow with twelve-inch turning radius on the centerline, and an assembly consisting of two sections of 8-in. tube joined at right angles by means of a section of turning vanes similar to the "Ducturn" presently used in the tunnel system. It was determined that the shortest possible shaft extension lengths were 14.915 in., 16.975 in., and 16.765 in. respectively. Since it was known that this length would be critical, it was decided to let it control the selection. The "short-radius" elbow is obviously the logical choice. Fig. 2 shows a development of the flow area which will be obtained through the "short-radius" elbow. It can be seen that pressure changes along the centerline will be regular and always in one direction, although highest acceleration occurs first on the under side of the nose cone, and subsequently on the upper side as the stream progresses through the elbow. This sequence may well be advantageous from the point of view of possible separation on the inside of the turn. At any rate, as has been mentioned previously, it is considered that any turbulence and other irregularities produced in the stream by the passage through the elbow will be obliterated sufficiently in the process of diffusion into the combustion chamber passages.

3.31 Appendix A presents an approximate analysis of the dynamics of the extension shaft required with the short-radius elbow. On the basis of an equivalent beam with pinned ends the critical speed would be 18,250 rpm. On the basis of the beam with fully fixed ends the critical speed would be 50,500 rpm. Although the maximum desired operating speed is almost identical with the former, the design is considered to be reasonably conservative in that the shaft extension is rigidly connected to other members at

both bearing supports, so that a considerable degree of fixity is obtained.

3.32 Appendix J shows that a tapered joint between the shaft extension and the dynamometer coupling is adequate for the applicable power transmission, so that costly splined joints can be eliminated.

3.33 The tunnel provides an airstream of controlled humidity. Further, the mass flow in the stream is normally measured at a point considerably downstream from the new elbow, and flow through the turbine must be known accurately. Therefore leakage into or out of the tunnel must be minimized, and for this reason the joint between the extension shaft and the tunnel elbow must be sealed as effectively as possible. This is accomplished by a labyrinth seal and gasket (Figs. 12 and 13). The seal is secured to the tunnel adapter pad (Fig. 11). Since the seal depends on very small clearances on the shaft for its effectiveness, and it is of significant length, the rigid specification that the inner face of the adapter pad be normal to the shaft axis, requiring final machining after assembly, is necessary. The securing bolt holes in the adapter have been specified oversize purposely to provide for proper radial positioning. Further, it was found that a simple enclosed radial ball bearing was adequate as a thrust bearing, but it is required that it be in very close alignment with the shaft. Hence the specification the adapter pad outer end be truly normal to the shaft axis. Although small discrepancies in axial position of the adapter face can be accommodated by shimming between the bearing spacer, (Fig. 14) and the adapter, such differences should be kept to a minimum, and therefore it has been specified that the outer face of the adapter also be machined after assembly into the elbow.

3.4 The low-speed branch of the wind tunnel structurally is essentially a large inverted "U" which is supported at its lower ends in the basement of the laboratory, with no support on the second floor where the test installation is to be made. Any elongation in the vertical section before the test installation must result in added displacement of the horizontal center line of the tunnel from the level of the second floor. This would preclude the possibility of mounting the dynamometer on the second floor independent of the tunnel structure, since it is imperative that the dynamometer and turbine shaft axes be in closest possible alignment due to the high operating speed. This may be accomplished most handily by mounting directly on the lower tunnel adapter flange at the top of the vertical section of the tunnel, securing with the flange bolts. This is shown schematically in Fig. 3. The dynamometer mount is essentially a table with angle-iron legs which straddle the 8-inch elbow and project down to and are secured to the adapter flange. The table must be so constructed that the dynamometer securing bolts may be located at will, to permit adjustment in angular alignment. Coincidence of the two shaft axes at the coupling is ensured by the coupling design.

3.41 The nicest installation would have provided a flexible and self-aligning coupling between the turbine and the dynamometer. Several Boston representatives of nationally-known makers of flexible couplings were contacted and queried about providing such couplings for this application. There were no such standard couplings usable at above 10,000 rpm., and reference to the engineering departments of the several companies yielded only negative results. All were reluctant to even advise relative to this

speed-power combination. The Fast's Coupling Department of the Koppers Company, Inc., offered a special coupling, their Dwg. No. CS-12520-4, at a cost of $98.00. The coupling was 4 5/8-in. long and 4 15/16-in. on the O. D. The external area provided thereby was considered to be unacceptable due to the probability of showing high unmeasurable windage losses which would be a large portion of the total output of the turbine at low power-high speed settings. Therefore it was decided to specify a rigid coupling similar to the one used by the manufacturer of the turbo-jet engine. A rigid coupling is also specified for the junction between the turbine shaft and shaft extension. Twelve fitted coupling bolts are available from the old turbine-compressor connection. Due to the fact that this installation will be run at less than one-half the design peak torque, six of these available bolts will suffice in each of the two rigid couplings, and the coupling parts which are to be manufactured locally have been designed accordingly.

3.5 The turbine of the test installation is a high-speed device which is susceptible to very high acceleration in the unloaded condition. On several occasions when Westinghouse test personnel experienced no-load run-aways with a turbine of approximately the same size and power output, due to failure of water flow through their dynamometer, accelerations were estimated to be at the rate of 5000-10000 rpm/sec. It is obvious that an automatic system of emergency shut-down must be provided. It would be inadequate to provide merely for a tunnel compressor cut-off, since its inertia is considerable and there would also be a large mass of air under pressure upstream of the turbine. It was decided to provide

a safety valve in the air stream as close to the turbine as possible. This can be accomplished nicely at the lower tunnel adapter flange, and is represented by Fig. 5 and shown on Fig. 4. The spring loaded flapper valve is held open during normal operation by a latch. When the turbine exceeds a preset limit, the latch is withdrawn by a solenoid which is energized through a governor on the dynamometer. Ref. (a) shows schematically one possible design for the governor and associated electrical circuits. It should be modified to eliminate the emergency fuel cut-off (not needed) and to provide a tell-tale which would indicate that electrical energy is available for the solenoid circuit when needed. Provisions also should be made to cut off power to the wind tunnel compressor simultaneously with triggering of the safety valve. A hand hole is provided in the lower tunnel adapter flange to facilitate re-setting the safety valve with minimum alteration to the existing tunnel structure.

4.0 INSTRUMENTATION

4.1 Static temperature and pressure must be known at any point where Reynolds' Number is to be computed, along with stream velocity. The latter can be determined by comparing stagnation and static pressures, both of which can be measured in an airstream with a sufficient degree of accuracy. However, static temperature is not measured readily with sufficient accuracy, and must be reduced from stagnation temperature by means of the relation

$$T(\text{stat.}) = T(\text{stag.}) \left(\frac{P(\text{stat})}{P(\text{stag})}\right)^{\frac{k-1}{k}}$$

assuming isentropic compression to stagnation conditions, where k is the ratio of the specific heats of the airstream. Stagnation temperature can

be measured with reasonable accuracy (ref. (e)). To summarize, stagnation temperature and stagnation and static pressure must be measured immediately ahead of the wheel, where Reynolds' Numbers are to be compared. Determination of stagnation temperature behind the turbine wheel together with stagnation temperature ahead, will provide a basis for computation of turbine output from the relation for an adiabatic process $W_x = \Delta h = c_p(\Delta T_0)$, where W_x is shaft work (see any reference on Flow Processes). This computation should be useful for comparison with output as measured by the dynamometer, therefore providing a basis for an estimation of combined losses between the turbine and the dynamometer, such as to bearings, seals, windage, etc. The various losses would not be separable. Measurement of either stagnation or static pressure behind the turbine will provide the necessary additional data for computation of work input to the turbine. Provisions have been made to measure these values (see Figs. 22 and 27), namely:

(a) Total temperature at one location ahead of the turbine nozzles (at one side);

(b) Total and static pressure at the same plane as above, at top and bottom of the flow annulus;

(c) Total (and static, if required) pressure at three locations (top, side, and bottom) aft of the turbine, approximately 3 1/2 inches downstream in the straight portion formed by the tail cone of the turbojet engine. In this distance there will have been no change in the annular area or diameter from that of the turbine wheel, yet the flow will have been given a chance to obtain some degree of uniformity, and vibration-type impulses from the blades of the turbine will have been at least partially damped.

(d) Total temperature at the same plane as for (c), at a location half-way between two of the pressure measurement points.

4.11 The instrument access holes necessary for the above have all been dimensioned for an I. D. of 5/8-in. so that an instrument head 3/8-in. long by 1/4-in. round or square can be inserted. Fig. 23 shows schematically the necessary mating bushings to support and provide a seal on the shanks of the instruments. These are designed to accommodate an O-ring packing to prevent leakage, yet permit angular and radial adjustment for traversing. The access provided will be sufficient to accommodate claw-type yaw meters or 3-hole yaw probes (ref. (c)), either of which could incorporate an additional tube for measurement of static pressure; Kiel-type total pressure probe heads (ref. (d)); or total temperature heads similar to some of those shown in ref. (e).

4.12 It is considered that only two measurements of total and static pressure ahead of the turbine should suffice, since there should be no swirl at that point, the only probable differences resulting from the short turning radius with possible separation in the 8-inch elbow ahead. Hence measurements of pressure are taken only at the top and bottom, between which the greatest differences should be apparent. Three points of pressure measurement aft of the turbine are provided, since it is believed that swirl components in the flow might make for non-uniform distribution, both radially and circumferentially. Further, measurements will be more difficult to make at this station if both total and static determinations are required, since the direction of flow must be found by yaw meter. It is suggested that only total pressure be measured at this location, using Kiel-type total pressure instruments which are largely insensitive to yaw. Since sufficient data

can be obtained from measurements ahead of the turbine to determine flow and Reynolds Number, static pressure behind would be redundant information.

4.13 The stagnation temperature in the tunnel is controllable, and the turbine inlet temperature should be maintained as close to ambient room temperature as possible, so that heat transfer to the surroundings from the stream will be minimized. Under these conditions it is believed that total temperature need only be measured at one circumferential location before and behind the turbine, since there is no apparent reason for circumferential variation at any given axial station.

4.14 No provisions have been made for external manipulation of the airflow instruments ahead of the turbine. It will be noted that the instrument lines are brought out through the turbine mounting flange, indicated on Fig. 3. It is predicted that it will suffice to mount the probes so that they will be on the mean radius of the turbine blades, and since essentially axial flow should be obtained at this point they could be so oriented and set. However, it is recommended that the pressure probes be of the yaw-meter type for use as a check on direction. If an error in orientation were to be made, the tunnel could be stopped and the probes re-oriented manually through the hand hold provided for this purpose in the turbine mounting flange. Once the proper orientation has been obtained it should not change for the various conditions of flow, since the volume flow will be essentially invariable. In the event that these provisions are proven to be inadequate, then additional access provisions can be made for this station similar to those let into the main tunnel wall in the transverse plane of the after-turbine station. The latter make it

possible to manipulate the probes from outside the tunnel while running, since it would be benificial to make radial traverses at the after station, and may prove to be profitable to determine stream directions at that station.

4.15 Torque and speed will be measurable at the dynamometer. A "strobo-tachometer" is available in the Gas Turbine Laboratory at M.I.T. which has been shown to give accuracy within 1 per cent, which is considered to be adequate for the intended use.

5.0 ASSEMBLY OF EQUIPMENT

5.1 When and as parts become available, the installation should be assembled in the general order following; (see ref. (h) for Westinghouse Parts identification.)

5.11 Install the after oil seal and gasket on the after end of the turbine bearings housing, Westinghouse Part No. 23F574-1. WARNING: Before this is done the turbine rotor and shaft can slide freely in the bearings housing, and careless handling may result in damaged bearings or rotor blades. It will be observed that production notes have been etched onto the turbine shaft at the place where the seal is to be effected. Any external projections of the upset metal should be carefully honed off before the seal installation. This honing should be kept to a minimum to prevent getting the shaft out of round at this point. Take all possible precautions to prevent grit or fragments from entering the bearings. The rubber gasket is brought into position from the shaft coupling end of the sub-assembly, requiring some stretching over projections in the process. (Do not attempt

to disassemble the turbine sub-assembly for gasket installation, due to the care required in re-assembly of shaft and coupling.) Then bring the sealing halves into position, locating the radial splits in the two pairs ninety degrees apart. Hold the sealing pieces very lightly in contact with the shaft while securing, to obtain minimum clearance with the shaft. The sealing crests have been made purposely sharp, to be worn off into clearance during the initial stages of running. Be sure that the gasket is centered on the shaft after final securing. Rotate the turbine wheel by hand to be sure too much interference has not been provided. Safety wire the securing screws.

5.12 Silver-solder the evacuation pipe adapter pads on to the turbine bearings support and drill and tap in place as shown on Fig. 17. Make a trial assembly of the evacuation pipes to be sure they do not project inside the I.D. of the bearings support. Provide for subsequent identification of mating parts. The pipes should be seven inches in length, of three-eighths inch standard pipe, belled at one end for clamped hose fittings, and having tapered pipe threads at the other end. Braze on the instrumentation access pads on the combustion chamber housing as shown in Fig. 22. Drill 37/64 inch hole through each and be sure that the hole is continuous into the airstream passage. Tap as specified on Fig. 22.

5.13 Ream out spark plug holes in the combustion chamber casing to an I.D. of 13/16 inch.

5.14 Install the diffuser tail cone and the combustion chamber inner liners (Figs. 15 and 16), ensuring that the various holes and cutouts

provided for the evacuation pipe installation are properly oriented. Then bring the combustion chamber housing and nozzle sub-assembly, and the diffusor and bearings support sub-assembly together, drawing up slowly with the securing screws, and secure. WARNING: The nozzle blades are very loosely fitted in their shrouds, and easily dropped out. Make sure that the nozzle inner shroud is brought into position on the bearings support end at the same rate the two sub-assemblies approach at the main securing flange. These sub-assemblies are indexed for proper orientation by two punch pricks on the mating flange of the combustion chamber housing and on one of the mounting lugs on the diffusor.

5.15 Secure the turbine shaft extension to the turbine, using six of the twelve bolts (WE Part.No. *18H276 line 1) which effected the coupling between the turbine shaft and the compressor shaft. The resulting assembly should then be checked for static and dynamic balance, and run-out at the far end of the shaft extension. Safety with cotter keys, which should be weighed with their respective bolts before-hand. All possible care should be taken to ensure that weights are disposed symmetrically, for balance of a rapidly-rotating part which will be operating near its critical speed.

5.16 Assemble the Turbine Sub-assembly into the Bearing Support and Diffusor assembly. The after sealing gasket, Fig. 19, is mushroomed into the I.D. of the Bearings Support during this step, which will require some force. This force should not be imposed on the turbine or its shaft, since this would likely damage the outermost ridge of the labyrinth seal. All required force should be exerted by pulling on the bearings housing through

the diffusor. Safety-wire the securing bolts after drawing up tight.

5.17 Install the evacuation pipes, and pack in the spark-plug holes with cotton wicking, to minimize air leakage ahead of the turbine.

5.18 Install the nose cone (Fig. 21). Use gasket compound similar to Permatex, Type A, sparingly on as many of the mating surfaces as possible, taking care not to allow any to flow into the labyrinth sealing grooves or onto the turbine extension shaft. Remove all external excess compound after assembly. Rotate the turbine by hand to ensure that too much interference between labyrinth seals and shaft does not exist.

5.19 Slip the tunnel seal and gasket, Figs. 12 and 13, on to the turbine extension, but do not slide them too far along the shaft. Then mount the turbine test assembly on the turbine support flange and secure. Pull the tunnel seal and gasket onto the tunnel adapter pad (Fig. 11) from outside, and secure. The screw holes in the pad have been drilled oversize on purpose, to permit proper radial location of the seal and gasket relative to the shaft.

5.20 Install the thrust bearing on the dynamometer coupling half (Fig. 20), shrink on the bearing retainer ring (Fig. 26). Position spacer (Fig. 14).

5.21 Attach coupling-and-bearing assembly to the shaft extension and slip thrust bearing retainer (Fig. 14) into place, as shown in Fig. 3. Locate the centers of the two positioning pins from the bearing support. Remove support, coupling and bearing, and spacer, drill and tap for and install positioning pins.

5.22 Make final installation of the coupling assembly, torquing the

retaining nut to 500 in-lbs. (It will be permissable to use a Stillson wrench on the stub end of the turbine rotor, but the resulting burrs should be dressed off carefully subsequently).

5.23 Feel out the resultant axial limits of the turbine rotor position (these will be small due to the presence of the after seal), and use split shims between the bearing spacer and the tunnel adapter pad to set the rotor half way between the limits. Rotate the turbine by hand to be sure that sufficient freedom exists for normal operation in the air stream.

5.24 Install the evacuation pipes and connect by hoses to a gas-tight access through the turbine mounting flange.

5.25 Install the tunnel safety valve on the lower tunnel adapter flange, as shown on Fig. 4. The safety valve springs are designed to be loaded by a turn of 70-deg. when the valve is in the closed position, and should be installed accordingly.

5.26 The remainder of the assembly procedure for getting the test installation into the tunnel is straightforward and may be seen clearly in Fig. 3.

6.0 OPERATING INSTRUCTIONS AND PRECAUTIONS

6.1 It is recommended that sand-bagging or equivalent protection be provided in the plane of the turbine wheel and that of the coupling between turbine shaft extension and dynamometer shaft before operation is attempted.

6.2 Starting

6.21 Calibrate and set the safety governor (Section 3.5) for activation at 20,000 rpm. Check the Primary Safety Mechanism. Be sure that the governor parts are working freely, without binding, and that the stand-by electrical circuit for the safety air valve tripping solenoid is energized (tell-tale light is on). Manipulate the governor so that the electrical circuit to the solenoid is closed, and be sure that the valve does in fact close. Reset, and close the hand-hole in the lower tunnel adapter flange.

6.22 Start oil-air mist evacuator, and establish pressure within the bearings support at a value lower than the lowest predicted static pressure at turbine inlet. Do not operate at any setting where this difference can not be maintained.

6.23 Start the oil pump for the oil air mist only after proper pressures have been set in accordance with 6.22. (This step might well be accomplished immediately <u>after</u> the turbine starts rotating, in step 6.24).

6.24 Set the water level control on the dynamometer for maximum torque (water chamber full).

6.25 Start the wind tunnel and establish desired flow.

6.26 Slowly reduce torque on the dynamometer to increase turbine speed to that desired. Watch carefully for vibrations.

6.27 Shut down tunnel immediately if safety solenoid circuit tell-tale light goes off, and re-establish the circuit before attempting further operation. <u>It is to be strongly emphasized that this is a high-speed mechanism which is capable of very rapid acceleration on run-away due to torque failure.</u>

6.3 Stopping

6.31 Increase dynamometer torque to maximum available.

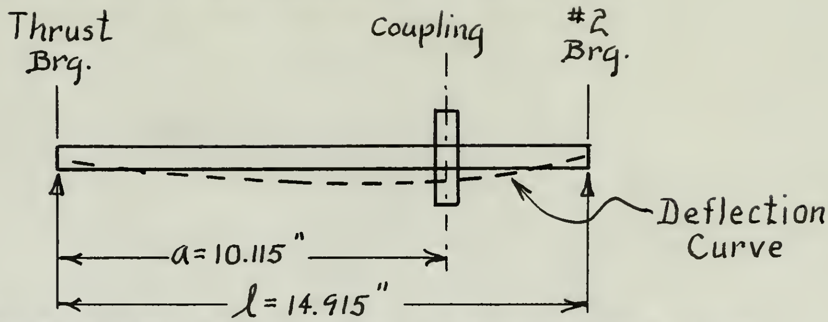
6.32 Stop the wind tunnel, thereby stopping the turbine.

6.33 After the turbine has stopped, shut off the oil pump. (This might well be accomplished slightly before turbine stops in step 6.32).

6.34 Let the oil-air mist evacuator run as long as possible after stopping the pump, to provide assurance that all oil has been removed from the turbine and associated system. It must be emphasized that the interior of the wind tunnel system must not be contaminated in any way. Take all possible precautions to ensure that any leakage between the test assembly and the tunnel air stream is always inward through the various seals and toward the evacuation lines.

APPENDIX A

DYNAMICS OF THE TURBINE SHAFT EXTENSION



The method of Rayleigh-Ritz for analysis of transversely vibrating beams will be followed. It will be remembered that, from the Law of Conservation of Energy, the maximum attainable potential energy of a vibrating system due to its position in space is equivalent to the maximum attainable kinetic energy of the system due to its motion, i.e.,

$$PE_{max} = KE_{max}$$

It will be assumed that the deflection curve for a vibrating pin-ended beam equivalent to the shaft system shown above can be represented by the equation

$$y = K \sin\left(\frac{\pi x}{\ell}\right) \cos\,(wt)$$

where K is some unknown constant

w is rotational velocity, rad./unit time

Then the slope at any point is

$$\frac{dy}{dx} = K\,\frac{\pi}{\ell}\,\cos\left(\frac{\pi x}{\ell}\right)\cos\,(wt) = \theta$$

and curvature at any point is

$$\frac{d^2y}{dx^2} = -K\,\frac{\pi^2}{\ell^2}\,\sin\left(\frac{\pi x}{\ell}\right)\cos\,(wt) = \frac{d\theta}{dx}$$

The velocity at any point is

$$\frac{dy}{dt} = -K\,w\,\sin\left(\frac{\pi x}{\ell}\right)\sin\,(wt)$$

For a beam increment subjected to a bending moment M, which results in an angular deflection $d\Theta$, the work done in bringing about the final condition of loading and position is

$$\Delta(PE) = M\ (ave) \times d\Theta = 1/2\ M d\Theta$$

From Beam Theory we have

$$M = EI \frac{d^2y}{dx^2} = EI \frac{d\Theta}{dx}$$

Then

$$\Delta(PE) = 1/2\ EI \frac{d\Theta}{dx} d\Theta = \frac{EI}{2}\left(\frac{d\Theta}{dx}\right)^2 dx$$

and

$$PE_{tot.} = \frac{EI}{2}\int_0^\ell \left(\frac{d\theta}{dx}\right)^2 dx = \frac{EI}{2}\int_0^\ell \left[-K\frac{\pi^2}{\ell^2}\cos(wt)\sin\left(\frac{\pi x}{\ell}\right)\right]^2 dx$$

$$= K^2\ \frac{\pi^4 EI}{2\ell^4}\cos^2(wt)\int_0^\ell \sin^2\left(\frac{\pi x}{\ell}\right)dx$$

$$= K^2\ \frac{\pi^4 EI}{4\ell^3}\cos^2(wt)$$

Incremental kinetic energy in the beam is $\frac{dm}{2} \times \left(\frac{dy}{dx}\right)^2$

and that of the coupling is $\frac{Mc}{2} \times \left(\frac{dyc}{dt}\right)^2$

where M_c is the mass of the coupling.

but $dm = \frac{\rho A dx}{g}$

where ρ is density, A is the sectional area

Then

$$KE_{tot} = K^2 \int_0^\ell \frac{\rho A}{2g} w^2 \sin^2(wt)\left[\sin^2\frac{\pi x}{\ell}\right]dx + \frac{M_c}{2}K^2 w^2 \sin^2(wt)\sin^2\frac{\pi a}{\ell}$$

$$= K^2\left[\frac{\rho A \ell}{4g}w^2 + \frac{M_c w^2}{2}\sin\left(\frac{\pi a}{\ell}\right)\right]\sin^2(wt)$$

Since

$$KE_{tot} = PE_{tot},$$

$$K^2\left[\frac{\rho A \ell}{4g} w^2 + \frac{M_c w^2}{2}\sin\left(\frac{\pi a}{\ell}\right)\right]\sin^2(wt) \quad = \quad K^2\,\frac{\pi^4 EI}{4\ell^3}\cos^2(wt)$$

To maximize, $\sin^2(wt) = \cos^2(wt) = 1$

(The maximum Potential Energy occurs at $\frac{\pi}{2}$ radians before and after the time where Kinetic Energy is maximum)

Therefore $$w^2\left[\frac{\rho A \ell}{4g} + \frac{M_c}{2}\sin^2\left(\frac{\pi a}{\ell}\right)\right] \quad = \quad \frac{\pi^4 EI}{4\ell^3}$$

and

$$w = \sqrt{\frac{\pi^4 EI}{\ell^3} \times \frac{1}{\frac{\rho A \ell}{g} + 2M_c \sin^2\left(\frac{\pi a}{\ell}\right)}}$$

For the steel shaft of diameter 1 1/8 in.,

$EI = 2.357 \times 10^6$	lb.-sq.in.
$\rho = 0.286$	lb./cu.in.
$a = 10.115$	in.
$l = 14.915$	in.
$g = 386$	in./sec^2

The coupling weight is 2.21 lb.

Solving the final equation above, W=1910 rad./sec. which is, then, the critical speed for the shaft extension.

and

$$N = \frac{w}{2\pi} \times 60 = \frac{1905}{2\pi} \times 60 = 18{,}200 \text{ rpm}$$

- -

For the case of the equivalent vibrating beam with rigidly fixed ends the equation of the deflection curve is assumed to be $y = K\left[1 - \cos\left(\frac{2\pi x}{\ell}\right)\right]\cos(wt)$

Proceeding as before $w = 5280$ rad./sec.

and $N = 50,500$ rpm.

These speeds represent the extremes between which it might be expected the shaft extension will have its critical speed. It is to be presumed that in the actual case there will be a considerable degree of fixity at both ends.

APPENDIX B

DETERMINATION OF TURBINE TEST OPERATING CONDITIONS

The well-known dimensionless parameters relating to speed and weight rate of flow in the turbine, for specification of performance, are:

$\frac{N}{\sqrt{\theta}}$, where N = rpm., θ is ratio of actual temperature to standard temperature (usually 520 deg. R) in absolute units

$\frac{w_a \sqrt{\theta}}{\delta}$, where w_a = weight rate of air flow (lbs./sec.)

θ is as above

δ is the ratio of actual static pressure to standard pressure (usually 14.7 psia or 29.92 in. Hg.)

Taking typical performance data from ref. (g) at rated power,

N = 34,000 rpm

T_4 = 1954 deg. R. where T_4 is stagnation temperature ahead of the Turbine

P_4 = 84.63 in. Hg where P_4 is stagnation pressure ahead.

w_a = 6.64 lbs./sec.

Tunnel Conditions for greatest weight flow can be maintained at 560 deg. R. and 1.6 atmospheres = 47.87 in.Hg., maximum desired.

Then, to obtain dynamic similarity between desired test conditions and design conditions

$$\left(\frac{N}{\sqrt{\theta}}\right)_{Test} = \left(\frac{N}{\sqrt{\theta}}\right)_{Design}$$

so

$$N_{Test} = N_{Design}\sqrt{\frac{\theta_{Test}}{\theta_{Design}}} = 34000\sqrt{\frac{560}{1954}} = 18,220 \text{ rpm}$$

and $$\left(\frac{W_a\sqrt{\theta}}{\delta}\right)_{TEST} = \left(\frac{W_a\sqrt{\theta}}{\delta}\right)_{DESIGN}$$

so $$W_{a_{TEST}} = W_{a_{DESIGN}} \times \frac{\delta_{TEST}}{\delta_{DESIGN}} \times \sqrt{\frac{\theta_{DESIGN}}{\theta_{TEST}}}$$

$$= 6.64 \times \frac{48.87}{84.63} \times \sqrt{\frac{1954}{560}}$$

$$W_{a_{TEST}} = 7.005 \text{ lbs./sec.}$$

APPENDIX C

DETERMINATION OF STATE OF AIR AT TURBINE INLET

Area of the Turbine Annulus:

Outer Diameter = 8.1 in.

Inner Diameter = 5.1 in.

$$A = \frac{\pi}{4}\left(\frac{D_o^2 - D_i^2}{144}\right) = \frac{\pi}{4}\left[\frac{(8.1)^2 - (5.1)^2}{144}\right] = 0.216 \text{ ft.}$$

Since only the stagnation conditions are pre-determined by the tunnel, static conditions must be obtained by trial and error. For first approximation use density based on stagnation conditions.

$\rho_o = \dfrac{P_o}{R \cdot T_o}$ from Gas Laws, where the subscript "o" signifies stagnation conditions.

$$= \frac{14.7 \times 1.6 \times 144}{53.35 \times 560} = 0.1132 \text{ lbs./cu.ft.}$$

Then $V = \dfrac{w_a}{A\rho}$ where V is stream velocity in ft./sec.

$$= \frac{7.005}{.216 \times .1132} = 286 \text{ ft./sec.}$$

Then $P = P_o - \dfrac{\rho V^2}{2g} = 1.6 \times 14.7 \times 144 - 1/2 \times \dfrac{.1132}{32.2} \times (286)^2$

$$= 3244 \text{ lbs./sq.ft.}$$

and $T = T_o - \dfrac{V^2}{2gJc_p}$ where the specific heat at constant pressure C_p = 0.2399 BTU/#/deg. F at 560 deg. R.

$$T = 560 - \frac{(286)^2}{64.4 \times 778 \times .2399} = 560 - 6.8 = 553.2$$

Then, for second trial, $\rho = \dfrac{3244}{53.35 \times 553.2} = .1099$ lbs./cu.ft.

$$V = \frac{7.005}{.216 \times .1099} = 295 \text{ ft./sec.}$$

$P = 3241$ lbs./sq.ft.   and $T = 553$ deg. R

The third trial shows

$= .1098$ lbs./cu.ft. $V = 295$ ft./sec.

$P = 3242$ lbs./sq.ft. $= 22.5$ lbs./sq.in.

$T = 553$ deg. R

APPENDIX D

DETERMINATION OF REYNOLDS' NUMBERS

Reynolds' Number for the required investigation will be based on state and velocity just ahead of the Turbine Nozzle Ring. The characteristic length will be the chord of the turbine blade at mean radius.

$N_R = \frac{\rho V l}{\mu}$, where N_R is Reynolds' Number

ρ is mass density (lbs-sec^2/ft^4)

V is velocity (ft/sec)

l is characteristic length (ft)

μ is viscosity (lbs-sec/sq ft)

$$\mu = \mu_0 \left(\frac{T_0 + C}{T + C}\right)\left(\frac{T}{T_0}\right)^{1.5}$$ from Ref. (i)

where C = 120 for air

T_0 = 273 deg. C.

$\mu_0 = 170.9 \times 10^{-6}$ poises

T = 553 x 5/9 = 308 deg. C.

so $= 170.9 \times 10^{-6}\left(\frac{273 + 120}{308 + 120}\right)\left(\frac{308}{273}\right)^{1.5} = 188.0 \times 10^{-6}$ poises

$= 188.0 \times 10^{-6} \times 2.088 \times 10^{-3} = 3.925 \times 10^{-7}$ lbs.-sec./sq.ft.

$$l = \frac{1.1875}{12} = 0.099 \text{ ft.}$$

Then $N_{R_{max}} = \dfrac{.1098 \times 295 \times 0.099}{32.2 \times 3.925 \times 10^{-7}} = 253{,}500.$

The tunnel can be evacuated to one-tenth of the maximum pressure used above, at the same turbine inlet temperature

Then $N_{R_{min}} = \frac{1}{10} \times N_{R_{max}} = 25{,}350$

APPENDIX E

DETERMINATION OF TURBINE POWER OUTPUT

$T_o = 560°$

P_o = 1.6 atmospheres = 1.6 x 14.7 = 23.53 psia.

Normal Rated Power is obtained at $\frac{P_i}{P_f} = 2$

where P_i is stagnation pressure at entrance)
) see Ref. (g)
P_f is stagnation pressure at exit)

at $T_{oi} = 560$ $H_i = 133.86$ $p_{r_i} = 1.5742$)
)
) from Gas Tables.
$Pr_f = \frac{1}{2} \times 1.5742 = .7871$ $h_f = 109.74$)

$\Delta h_s = 133.86 - 109.74 = 24.12$ BTU/lb.

$HP = \frac{24.12 \times 778}{550} W_a = 34.1 W_a$ assuming $\eta_t = 100$

Therefore maximum possible power at desired operation

= 34.1 x 7.005 = 239.5 hp.

APPENDIX F

ANALYSIS OF THE CHARACTERISTICS OF A FLAT DISC RUNNING IN WATER

$$H.P. = \frac{4\pi f \ w^{n+1}}{(n+3) \times 550} R_o^{n+3} - R_i^{n+3} \qquad \text{(see ref (f))}$$

where

f = coefficient of friction

w = angular velocity, radians/sec.

R_o = outer radius of disc, ft.

R_i = inner radius of water level, ft.

Experimental results show $n = 1.8$, $f = .004$ for rough surfaces, .002 for polished surfaces.

$$R_o^{4.8} - R_i^{4.8} = \frac{H.P. \times 4.8 \times 550}{4\pi \times .004 \times w^{2.8}}$$

Design conditions are

N = 18,220 rpm

H.P. = 239.5

Therefore it is desired to design a brake which will absorb 250 H.P. and over a range of $\frac{3}{4}$ design speed to maximum allowable.

$\frac{3}{4}N$ = 13,680 rpm

Maximum allowable = 20,000 rpm

It can be seen from the foregoing equation that the slowest speed will determine the largest radius. Therefore for 250 H.P. and 13,680 rpm

$$R_o^{4.8} - R_i^{4.8} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times \left(\frac{2\pi}{60} \times 13,680\right)^{2.8}} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times 6.82 \times 10^8}$$

$$R_o^{4.8} - R_i^{4.8} = .01929$$

letting $R_i = 0$ for trial radius

$$R_o = \sqrt[4.8]{.01929} = .439 \text{ ft.} = 5.26 \text{ in.}$$

Now set $R_o = 5.4$ in. $= .45$ ft.

$$R_i = \sqrt[4.8]{(.45)^{4.8} - .01929} = \sqrt[4.8]{.0218 - .01929} = .288 \text{ ft.} = 3.46 \text{ in.}$$

for the lowest power output H.P. = 25

$$R_o^{4.8} - R_i^{4.8} = .001929$$

Under this condition the inner radius of the water level would be

$$R_i = \sqrt[4.8]{.0218 - .0001929} = .441 \text{ ft.} = 5.292 \text{ in.}$$

For the maximum allowable speed and 250 H.P.

$w = 2093$

$w^{2.8} = 1.991 \times 10^9$

$$R_o^{4.8} - R_i^{4.8} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times 1.991 \times 10^9} = .00659$$

$$R_i = \sqrt[4.8]{.0218 - .00659} = .418 \text{ ft.} = 5.02 \text{ in.}$$

for lowest power output at 20,000 rpm

$$R_i = \sqrt[4.8]{.0218 - .000659} = .4475 \text{ ft.} = 5.37 \text{ in.}$$

Compare this to the O.D. of the disc = 5.40. Instability is highly probable!!!

Water flow required for cooling purposes

Let allowable temperature rise = 100 F. degrees

$$\text{Heat in.} = \frac{\text{h.p.} \times 33000}{J} = \frac{250 \times 33000}{778} = 10{,}600 \text{ BTU/min.}$$

wt. = q/ T = 10,600/100 = 106 lbs. of water/min.

gal./min. = 106/8.345 = 12.71

It can be readily seen with this rate of flow and the narrow range of water level, that speed control would not be accurate enough for use.

APPENDIX G

ANALYSIS OF THE CHARACTERISTICS OF A CYLINDER PARTIALLY SUBMERGED IN WATER

Assume that no end face is exposed to water

Then $T = 2\pi f l\ w^n\ R^{n+2}$ (see Ref. (f))

where T is torque on cylinder (ft-lb)

f is coefficient of friction = 0.004

l is length of the cylinder (ft)

w is rotational speed, rad/sec

R is the external radius (ft)

n = 1.8

And Horsepower = HP = T w/550

$$= \frac{2\pi f\ l\ w^{n+1}\ R^{n+2}}{550}$$

$$\text{so } l = \frac{HP \times 550}{2\pi f\ w^{2.8}\ R^{3.8}}$$ (from Appendix F, $w^{2.8} = 1.991 \times 10^9$)

Then, for 250 hp. at 20,000 rpm,

$$l = 0.00275\ (\tfrac{1}{R})^{3.8}$$

Assume R = 2 in.

$$\text{Then } l = .00275(\tfrac{12}{2})^{3.8} = 2.475 \text{ ft.} = 29.7 \text{ in.}$$

Power varies directly with length. For one-tenth of maximum desired operating output (=23.95 hp)

$$l = \frac{23.95}{250} \times 29.7 = 2.85 \text{ in.}$$

- -

Calculation of pressure forces required to balance centrifugal forces of the water rotating in the housing.

Centrifugal force of an element of water of unit depth

$$= \frac{\rho}{g} dr \; r \, d\theta \, \frac{V_r^2}{r}$$

where ρ = density (lbs./cu.ft.)

g = gravitational constant (32.2 ft./sec.^{2})

V_r = linear velocity at radius r (ft./sec.)

r = radial position of the element (ft.)

dr = radial length of the element

$rd\theta$ = the circumferential length

The balancing pressure force (centripetal force)

$$= \rho h \cdot r d\theta$$

where h is the hydrostatic head (ft.)

other symbols as above.

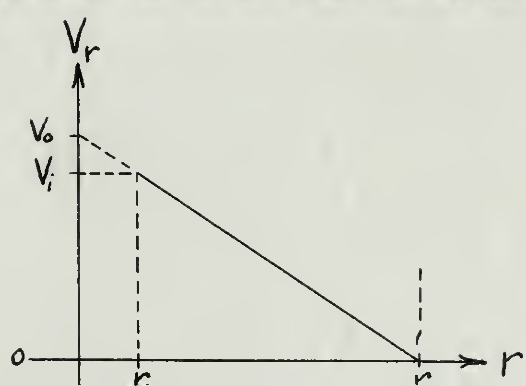
$$\text{Then } \rho h \; r d\theta = \frac{\rho}{g} \frac{V_r^2}{r} \, dr \; r d\theta$$

$$\text{and } h_{max} = \int_{r_i}^{r_c} \frac{1}{g} \frac{V_r^2}{r} \, dr$$

where r_c is the inner radius at the casing

r_i is the outer radius of the cylinder.

Now assume velocity distribution from cylinder to casing is linear, coming to zero at the casing, and having zero slip on the cylinder, (shown herewith):



V_i = velocity on surface of brake cylinder

$$= wr_i = \frac{20000}{60} \times 2\pi \times \frac{2}{12}$$

$$= 348 \text{ ft./sec.}$$

Now $\dfrac{V_i}{r_c - r_i} = \dfrac{V_o}{r_c}$ and $V_o = \dfrac{348\ r_c}{r_c - \frac{2}{12}} = \dfrac{348\ r_c}{r_c - .1667}$

and $\dfrac{r}{r_c} + \dfrac{V_r}{V_o} = 1$

$$\text{so } V_r = \frac{r_c - r}{r_c}\ V_o = \frac{348\ (r_c - r)}{r_c - .1667}$$

$$\text{Then } h_{max} = \frac{348^2}{(r_c - .1667)^2} \times \frac{1}{g} \times \int_{.1667}^{r_c} \left(\frac{r_c^2}{r} - 2\ r_c + r\right) dr$$

so, when $r_c = 1$ ft., $h_{max} = 3300$ ft.

or 1430 lbs./in.^{2}

It can be seen from the nature of the derived formulation for h that it will decrease as r_c increases. However, if the enclosing casing is to be kept within reasonable dimensional limits, an enclosed pressure tight casing will be required, resulting in considerable unpredictable circulation. At low power settings with high speed there would be great likelihood of instability due to the short submerged depth on the brake cylinder. It would be impractical to decrease the cylinder diameter much under 4 in., because of the large submerged depth that would be required at full power.

APPENDIX H

DETERMINATION OF POWER ABSORPTION IN A CENTRIFUGAL WATER PUMP

Assume that the water enters axially at the center and leaves tangentially at the periphery. Then power absorption is that due to the change in tangential momentum imposed on the water

$$HP = \frac{1}{2} MV^2/33000 = \frac{1}{2}\frac{W}{g} \; V^2/33000$$

where W is the weight rate of water flow (lbs./min.)

g is the gravitational constant (32.2 ft./sec.^{2})

V is the linear velocity at the wheel rim (ft./sec.)

and $W = \dfrac{2 \times HP \times g \times 33000}{V^2}$

taking the density of water as 8 lbs./gal.

$$Q = \frac{2 \times HP \times g \times 33000}{8 \times V^2} = \frac{2 \times HP \times g \times 33000}{8 \times (wr)^2}$$

where Q is volume flow (gal./min.)

w is the rotational speed (rad./sec.)

r is the rim radius (ft.)

Then, for 250 hp. at 20000 rpm

$$Q = \frac{2 \times 250 \times 32.2 \times 33000}{8 \times (2\pi \times \frac{20000}{60})^2 \; r^2} = 15.22 \times \frac{1}{r^2}$$

also $V = wr = 2\pi \times \dfrac{20000}{60} \times r = 2092 \; r$

The following solutions are found:

r (ft.)	wr = V (ft./sec.)	Q (gal./min.)
0.5	1046	60.3
0.4	836	95.2
0.3	627	169.0

It can be seen that, for a reasonable flow, wheel diameters and resulting exit velocities are excessively high (see Section 3.13).

APPENDIX I

THRUST BEARING DESIGN

Assume a 50% Reaction Turbine

Then, when $\frac{(P_i)}{(P_f)\text{Design}} = 2,$ $\qquad \frac{P_{i_r}}{P_f} = \sqrt{2}$

where P_i is stagnation pressure ahead of nozzles

P_f is stagnation pressure behind rotor

P_{i_r} is stagnation pressure ahead of rotor

Maximum desired operating pressure $P_i = 1.6$ atmos.

$$\text{Then } P_{i_r} = \frac{1.6 \times 14.7}{\sqrt{2}} = 16.62 \text{ psia}$$

$$P_f = \frac{1.6 \times 14.7}{2} = 11.78 \text{ psia}$$

$$\Delta p = 4.84 \text{ psia}$$

Maximum Thrust $= A \times \Delta P$

where A is turbine annulus area = .216 sq.ft.

$T = .216 \times 144 \times 4.84 = 150.5$ lbs.

Consider Norma-Hoffman Double Plate Single Row Inch-Type Bearing

Medium size, Series 307-PP

Rated Load at 3000 rpm = 820 lbs.

Rated Load at 20000 rpm = 480 lbs., by extrapolation of tabulated values.

The manufacturer's catalog gives

$P = 0.5\ R + 1.5\ T$

Where P is equivalent load for rating

R is radial Load

T is Thrust Load

In the desired application $R = 0$

so $P = 1.5 \times 150.5 = 226$ lbs.

Therefore Bearing factor of safety $= \frac{480}{226} = 2.2$

(based on bearing life of 10000 hours, but extrapolating beyond maximum tabulated speed rating.)

- -

Adequacy of Shrunk on Retainer Ring, Part No. D-1283.

When the tunnel is suddenly shut down, as when the air safety valve is triggered, the turbine becomes a compressor of low efficiency. This presumably happens when the load on the dynamometer is lost, and the turbine continues to run under its own inertia for a short period of time. Under these conditions the axial load on the turbine rotor reverses. Hence a shrunk on retainer ring will be required behind the bearing, on the coupling.

Use a steel ring with cross section 1/8 in. by 1/4 in. (see Fig. 26), with a Shrink Fit (Class 8) on the coupling where the O.D. is 1.3779 in.
Minimum interference is 0.0008 in., diametral

Now $e = \frac{Pl}{AE}$, where e = circumferential elongation in the ring

P = the tensile load in the ring

l = the mean circumference of the ring

A is sectional area

E is modulus of elasticity.

$= \pi(\Delta d)$, where Δd is the interference

$$\text{Then } P = \frac{eAE}{l} = \frac{\pi \times \Delta d \times A \times E}{\pi \times d_m} = \frac{.0008 \times \frac{1}{4} \times \frac{1}{8} \times 30 \times 10^6}{3/2}$$

$= 500$ lbs.

Now $2P = 2\,p\,r\,w$

where p is normal pressure on the inner surface of the ring

r is the inner radius of the ring

w is the axial width of the ring

so $p = \dfrac{P}{rw}$

Total Normal Load $= N = p \times 2\pi r \times w = 2\pi P$

$N = 3140$ lbs.

Allowable axial load $= N \times f$

where f is static coef. of friction (0.15 for steel to steel)

Allowable axial load $= 0.15 \times 3140 = 471$ lbs.

Assume that the turbine can provide a pressure ratio of 2/1 when acting as a pump (very conservative, since the blades will probably be stalled, and efficiency will be low).

Also assume that the tunnel compressor is working on a 2 to 1 pressure ratio, compressing to 1.6 atmospheres (the highest desired operating condition).

Then exit pressure on turbine $= p_e = 1/2 \times 1.6 \times 14.7 = 0.8 \times 14.7$

and inlet pressure on turbine $= p_i = (1/2)\,p_e = 0.4 \times 14.7$

Then total axial pressure force on rotor $= (0.8 - 0.4) \times 14.7 \times 144 \times .216$

$= 183$ lbs.

Factor of Safety provided by shrink ring

$= \dfrac{471}{183} = 2.58$

APPENDIX J

DETERMINATION OF TORQUE CAPACITY OF THE PLAIN TAPERED JOINT BETWEEN COUPLING AND SHAFT EXTENSION

Torque in Shaft = HP/w where HP is turbine power output

w is turbine rotational speed

Torque = $\frac{250 \times 33000}{20000 \times 2\pi}$ = 65.8 ft.-lbs. = 789 in.-lbs.

From ref.(I), static coef. friction, steel to steel, = 0.15

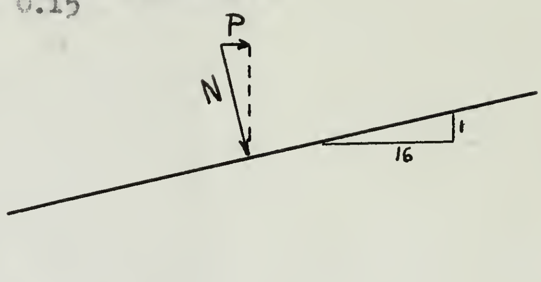
Radial Taper in Joint = 1/16 in/in

Average diam. in Joint = 1 1/16

Then total Surface force on Joint = $\frac{\text{Torque}}{\text{radius}}$

$= \frac{789}{17/32} = 1485$ lbs.

and total Normal force = $N = \frac{1485}{f} = \frac{1485}{0.15} = 9{,}900$ lbs.



$\frac{P}{N} = \frac{1}{\sqrt{1^2+16^2}} = \frac{1}{\sqrt{257}} = 0.0606$

so $P = 0.0606 \times 9{,}900 = 600$ lbs.

where P is the necessary axial force which must exist between coupling and shaft while running at full power.

The axial force on the turbine rotor tends to separate the two, so this also must be provided by the required retaining nut, i.e.

Axial force imposed by retaining nut = P = 600 + 150.5 = 750.5 lbs.

Using a 5/8-in. screw, National Fine Thread Series (18thds./in.),

Normal Force on Threads = $N_s = \frac{P}{\cos 30} = 866$ lbs.

Torque on Threads = $N_s \times f \times r_t = T_t$

where f is coef. of friction

r_t is pitch radius of thread

Use f for static friction, to be conservative

Then $T_t = 866 \times 0.15 \times \frac{0.59}{2} = 38.4$ lb.-in.

Assemble the coupling with 500 lb.-in. imposed on retaining nut.

Factor of Safety at Full Load = $\frac{500}{38.4} = 13.0$

It is recommended that high torque be used in setting up the assembly, to provide assurance that the coupling and shaft are properly aligned.

APPENDIX K

AIR SAFETY VALVE SPRING DESIGN

Fig. 5 shows the desired valve.

O.D. = 9 in.   Thickness = 1/4 in.

c.g. to hinge-line -- 5 in.

Material -- Aluminum Alloy, 24 ST

Density = 0.10 lbs./cu.in.

$$\text{Wt. of Valve} = \frac{\pi}{4} \times D^2 \times t \times \rho = \frac{\pi}{4} \times 81 \times \frac{1}{4} \times 0.10$$

$$= 1.6 \text{ lbs.}$$

Necessary seating torque about hinge = 1.6 x 4.82 = 7.72 lb.-in.

Use two springs, one left and one right hand, 0.093 in.diam. hard drawn spring wire on 0.625 mean spring diam., six active turns each.

From ref. (j), $T = \frac{Ed^4n}{10.8\ ND}$

where E is modulus of Elasticity

d is spring wire diam.

n is angle of deflection (no. turns)

N is no. of active turns

D is mean diam. of the spring

T is the spring torque

Take spring deflection of 1/2 turn for valve-open position

$$\text{Then } T = \frac{30 \times 10^6 \times (.093)^4 \times 1/2}{10.8 \times 6 \times 0.625} = 27.8 \text{ lb.-in. each}$$

Angle of turn when valve is seated is 70 deg.

Then $T_{seat.} = \frac{70}{180} \times 27.8 \times 2 = 21.6$ in lbs. seating torque

Factor of Safety when seated

$= \frac{21.6}{7.72} = 2.8$, and valve will stay safely closed without a latch.

See Fig. 7 for spring details

- -

Analysis of Closing Time of Air safety valve:

$$\alpha = \frac{d^2\theta}{dt^2} = \frac{T_\theta}{I}$$

where α = angular acceleration

θ = angle of valve, referenced to zero at open position (radians)

T_θ = torque on the valve, a function of θ

I = Mass moment of Inertia of the valve about the hinge.

When valve is open, $T_o = 55.6$ lb.-in.

When valve is closed, $T_c = 21.6 - 7.72$

$= 13.88$ lb. in.

$$\frac{\theta_T}{55.6} = \frac{110/57.3}{55.6 - 13.88}, \text{ so } \theta_T = \frac{55.6}{41.72} \times \frac{110}{57.3} = 2.56 \text{ rad.}$$

Then $T_\theta = 55.6 \left(1 - \frac{\theta}{\theta_T}\right) = 55.6 - 21.7\,\theta$

so $\frac{d^2\theta}{dt^2} = \frac{55.6 - 21.7\,\theta}{I}$

and $\frac{d^2\theta}{dt^2} + \frac{21.7}{I}\theta = \frac{55.6}{I}$

One possible solution is

$$\theta = C_1 \cos\sqrt{\frac{21.7}{I}}\,t + C_2 \sin\sqrt{\frac{21.7}{I}}\,t + \frac{55.6}{21.7}$$

$t = 0$ when $\theta = 0$ and $w = \frac{d\theta}{dt}$ when $\theta = 0$

Then $C_1 = -\frac{55.6}{21.7}$ and $C_2 = 0$

$$\text{so } \theta = \frac{55.6}{21.7}\left[1 - \cos\sqrt{\frac{21.7}{I}}\,t\right]$$

When the valve is closed, $\theta_c = \frac{110}{57.3}$ rad.

$$\text{so } \frac{110}{57.3} = \frac{55.6}{21.7}\left[1 - \cos\sqrt{\frac{21.7}{I}}\,t\right]$$

$$1 - \cos\sqrt{\frac{21.7}{I}}\,t = .750$$

$$\text{Now } I = \frac{W}{g}\left(\frac{D^2}{16} + d^2\right)$$

where W is weight of the valve in lbs.

g is gravitational constant (386 in./sec^2)

D is the valve diameter (9 in.)

d is distance from the c.g. of the valve to the hinge line (4.82 in.)

$$I = \frac{1.6}{386}\left(\frac{81}{16} + 23.25\right) = .1172 \text{ lbs.-in.-sec.}^2$$

$$\cos\sqrt{\frac{21.7}{.1172}} = \cos\sqrt{185} = \cos\,(13.6 \text{ rad.}) = .515$$

so the required time for closing $= t = \frac{1 - .750}{.515}$

$$t = 0.485 \text{ sec.}$$

It is to be noted that this represents the <u>minimum</u> closing time, since the air stream will provide considerable additional closing effect.

References: (a) "The Performance of Axial-Flow Turbines", D. G. Ainley, B.Sc., The Institution of Mechanical Engineers, Proceedings 1948, Vol. 159, War Emergency Issue No. 41, London.

(b) "An Investigation of the Properties of Eddy Current Brakes", Lof, J.L.C., Master's Thesis 1941, MIT.

(c) "Report on Characteristics of a 3 Hole Yaw Probe", Vance D. Baker, July 10, 1946, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).

(d) "Report on Kiel Probes", Marston Moffatt, October 17, 1945, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).

(e) "Report on Cylindrical Temperature Probes", Marston Moffatt, May 18, 1945, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).

(f) "Hydraulics and Its Applications", A.H. Gibson, Third Edition, 1925, D. Van Nostrand Company, New York, N.Y.

(g) "Turbo-Jet Aviation Log Book, Westinghouse Engine No. 5A1314-10". (Incustody of the Gas Turbine Library, MIT).

(h) "Handbook of Service Instructions for Turbo-Jet Engines," Model XJ32-WE-4 (X9.5B). (In custody of Gas Turbine Library, MIT).

(i) "Handbook of Engineering Fundamentals" O. W. Eshbach, 11th Printing, 1947, John Wiley & Sons, Inc. New York, N. Y.

(j) "Mechanical Spring Design Catalog", Wallace Barnes Co., Bristol, Conn.

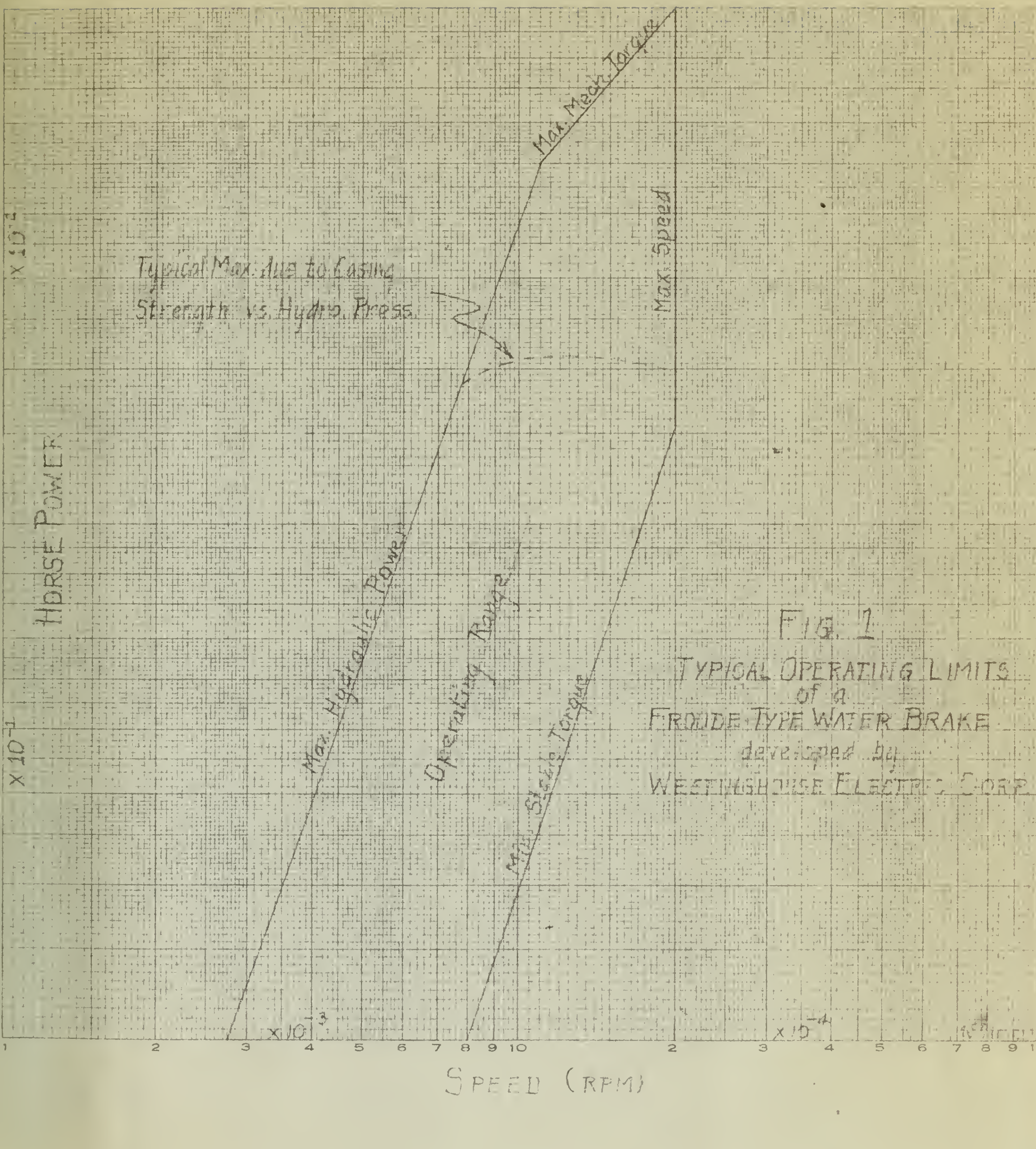


FIG. 1

TYPICAL OPERATING LIMITS of a FROUDE TYPE WATER BRAKE developed by WESTINGHOUSE ELECTRIC CORP.

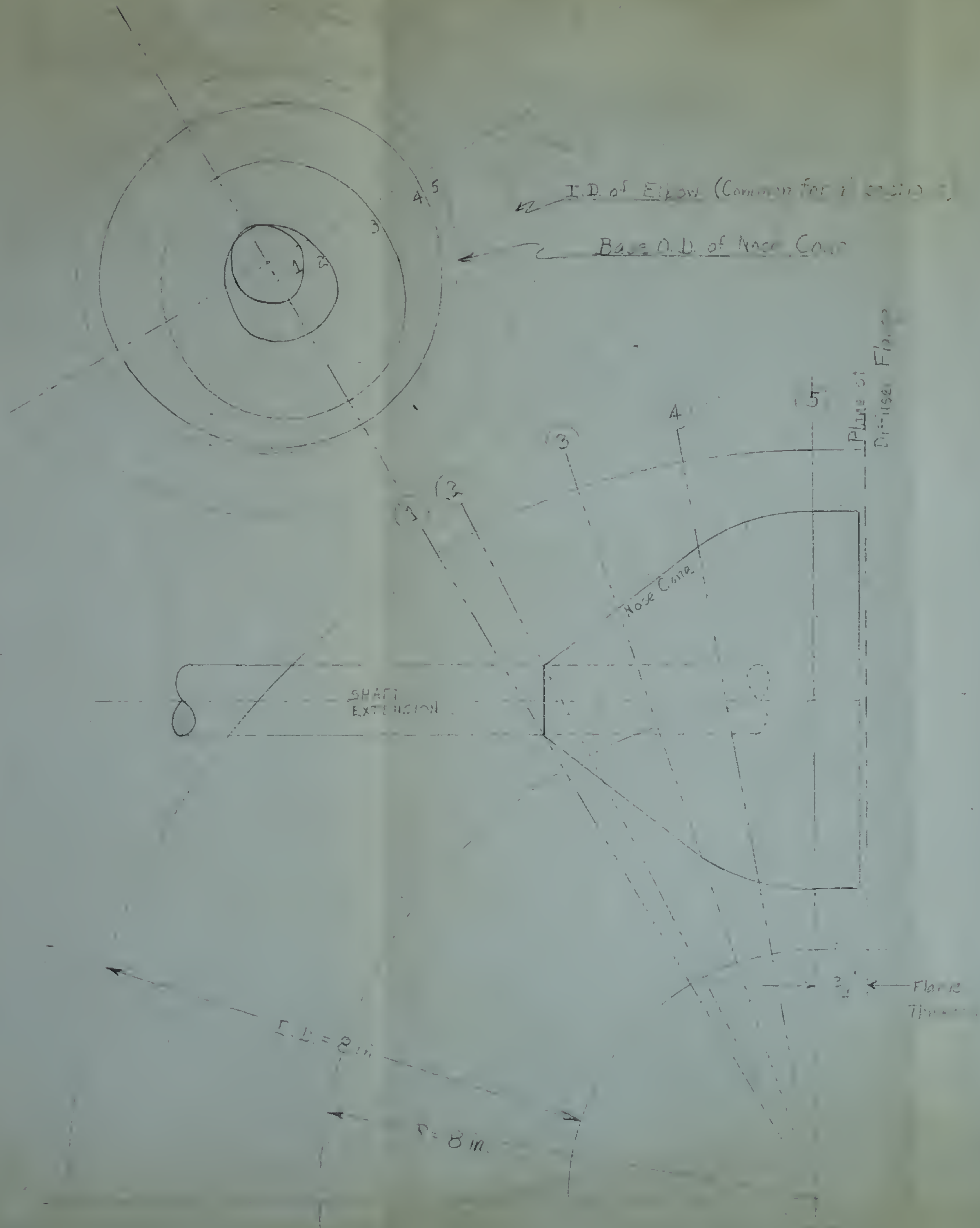
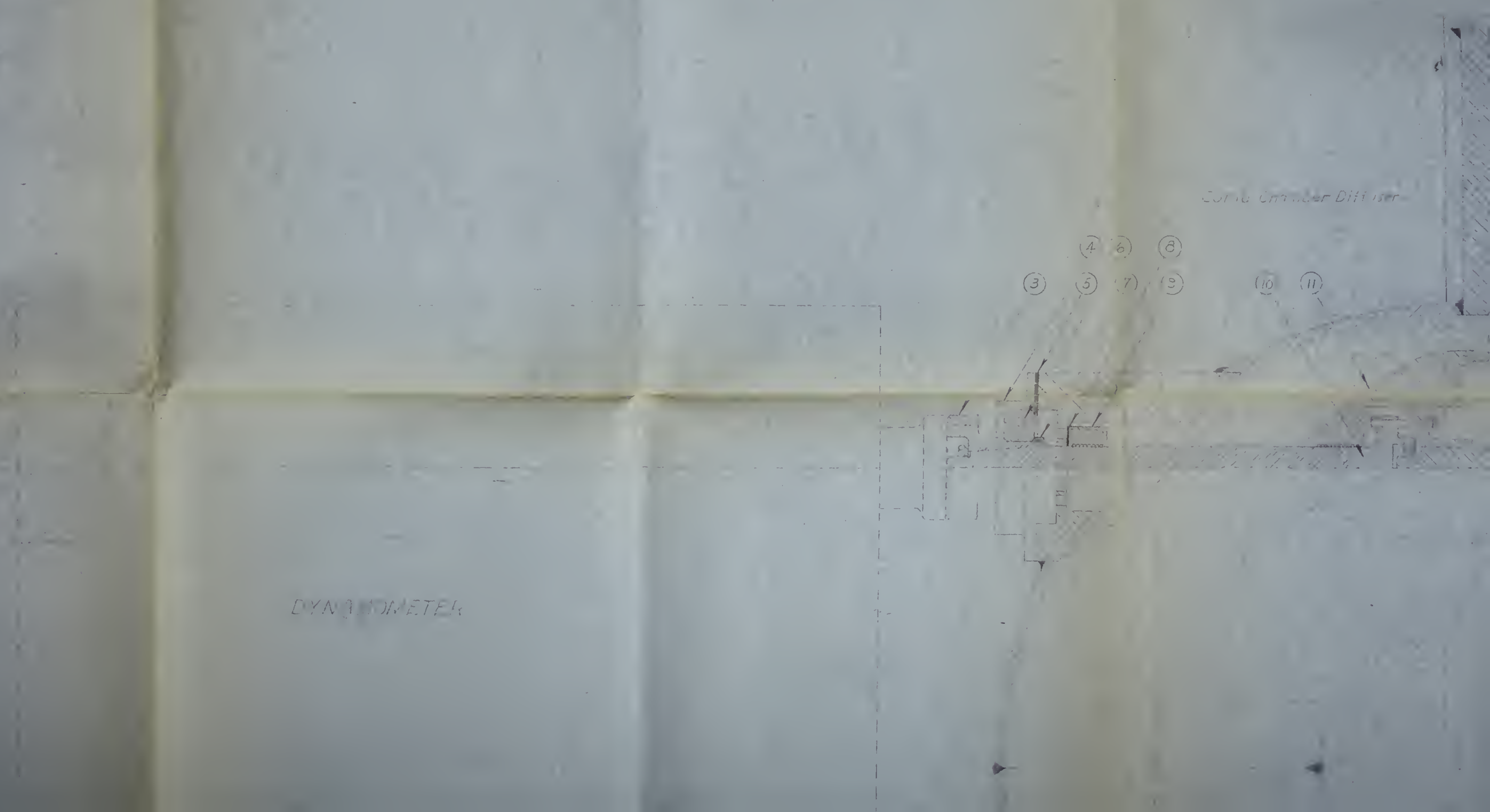


Fig. 2

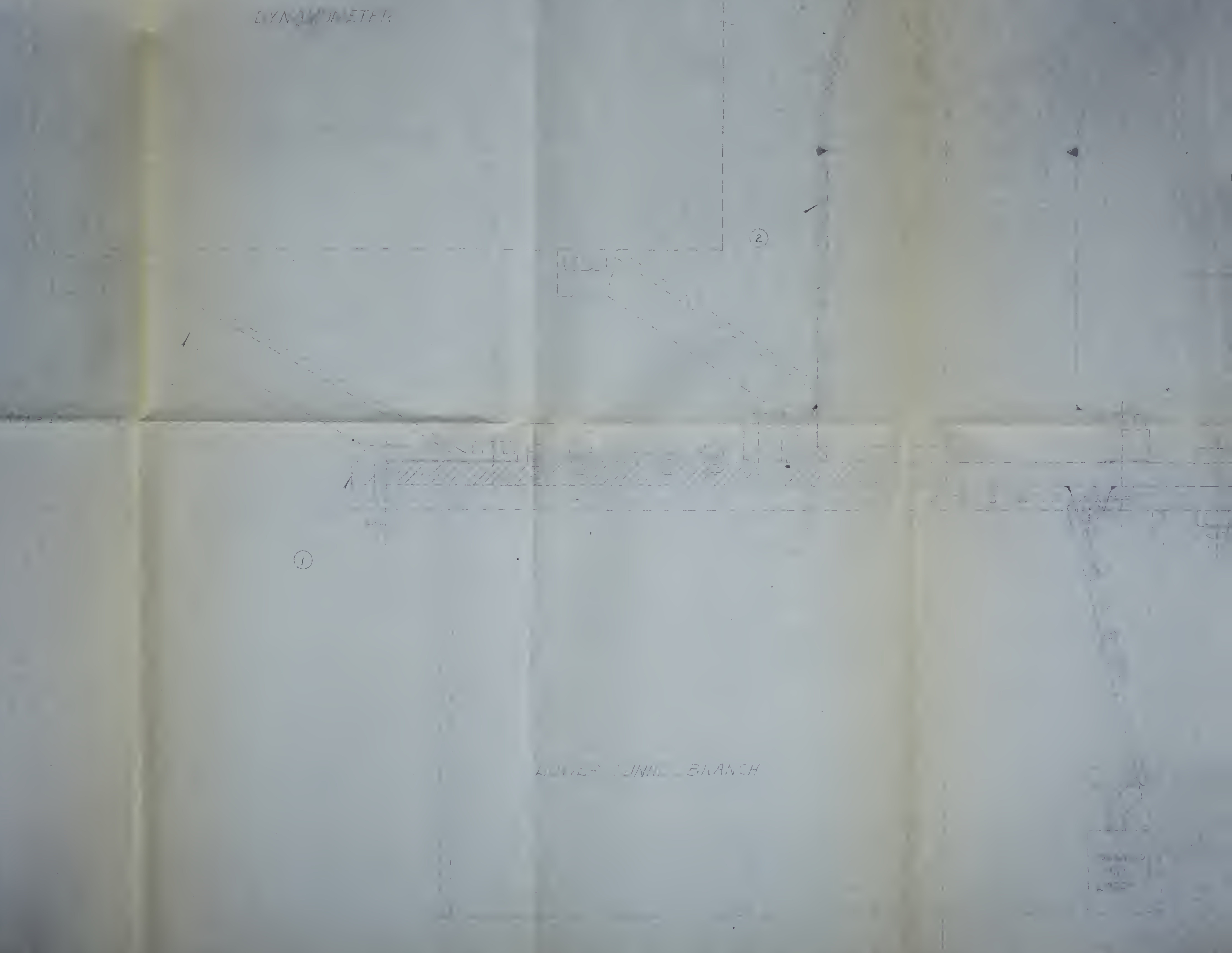
Sectional Shapes of Stream Passages Through 8 in "Short Radius" Elbow

BILL OF MATERIAL

		NAME	
1	[illegible]	[illegible] ADAPTER AND [illegible]	[illegible]
[illegible]	[illegible]	[illegible]	1
[illegible]	[illegible]	[illegible] EXTENSION [illegible]	1
4	[illegible]	BEARING RETAINER	1
[illegible]	[illegible]	SPACER	1
[illegible]		[illegible] NG ([illegible])	1
[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible] SHAFT [illegible] SEAL GASKET	1
[illegible]	[illegible]	[illegible] SEAL	1
[illegible]	[illegible]	SHAFT EXTENSION	1
[illegible]	[illegible]	[illegible]	1
[illegible]		EVACUATION PIPE	2
13	[illegible]	EVACUATION [illegible] ADAPTER	2
14	[illegible]	[illegible]	1
[illegible]	[illegible]	[illegible]	1
16	[illegible]	INNER [illegible] RING	1
17	[illegible]	[illegible] BUSHING	11
18	[illegible]	[illegible] PAD (COMP. [illegible])	2
[illegible]	[illegible]	[illegible]	1
[illegible]	[illegible]	[illegible]	1
21	[illegible] 91	[illegible]	4



17
12
13
14
15
16
17
18
19
20
17
21
10
11



DYNAMOMETER
2
1
BRANCH

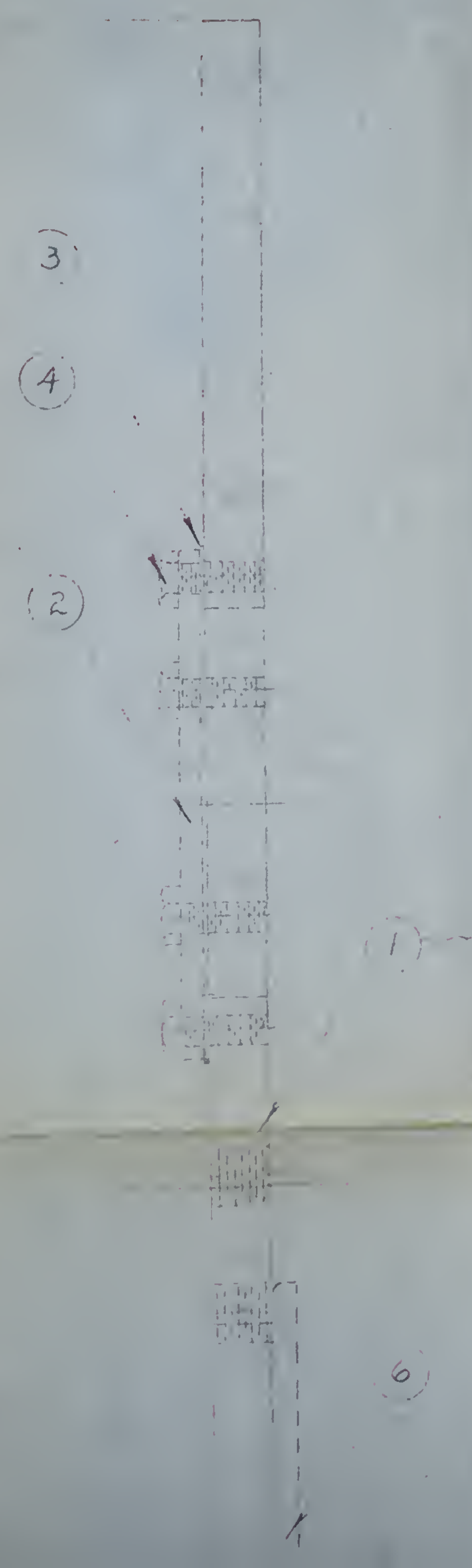

UPPER TUNNEL BRANCH
NO.
MATERIAL
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY
DRAWN BY
DATE 5/13/49
CHECKED BY
SCALE Half size
0-5259

FIG. 3

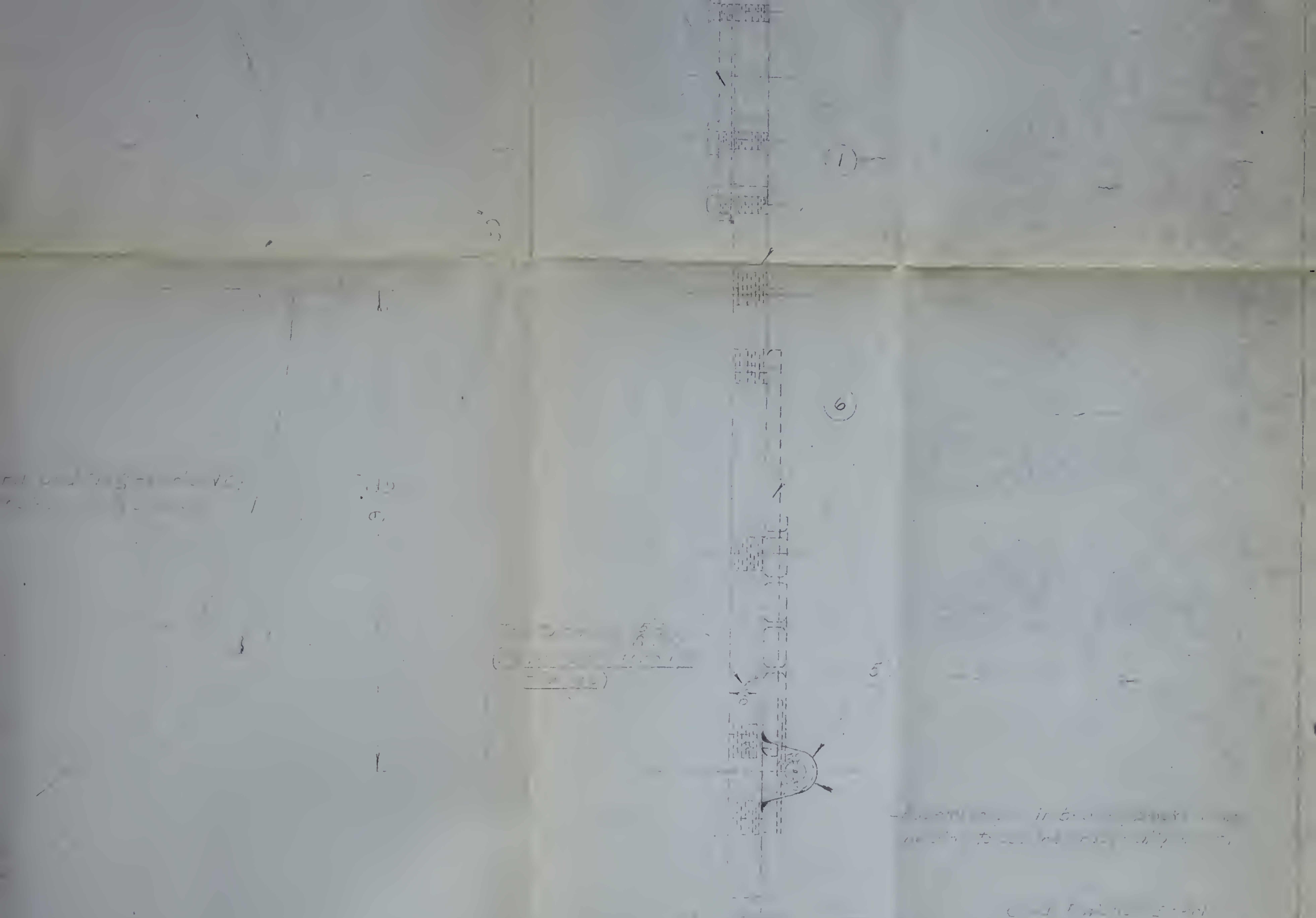

5" dia. hand hole
circle
9"
4

BILL OF MATERIAL.

	[illegible] NO.	NAME	[illegible]
1	O-4260	Flange	1
2	0-1265	Hand Hole Cover	
3		$\frac{1}{16}$ [illegible] Gasket	
4	$\frac{3}{8}$"-16 x $\frac{5}{8}$ lg.	[illegible] Screw	6
5	0-1262	Bearing Block.	2
6	0-2261	Safety Valve Ass'y	1
7	0-12[illegible]-1	[illegible] L.H.	1
8	0-12[illegible]-2	[illegible] R.H.	1
9	0-1264	Hinge Pin	1
10		Cotter Pin	[illegible]



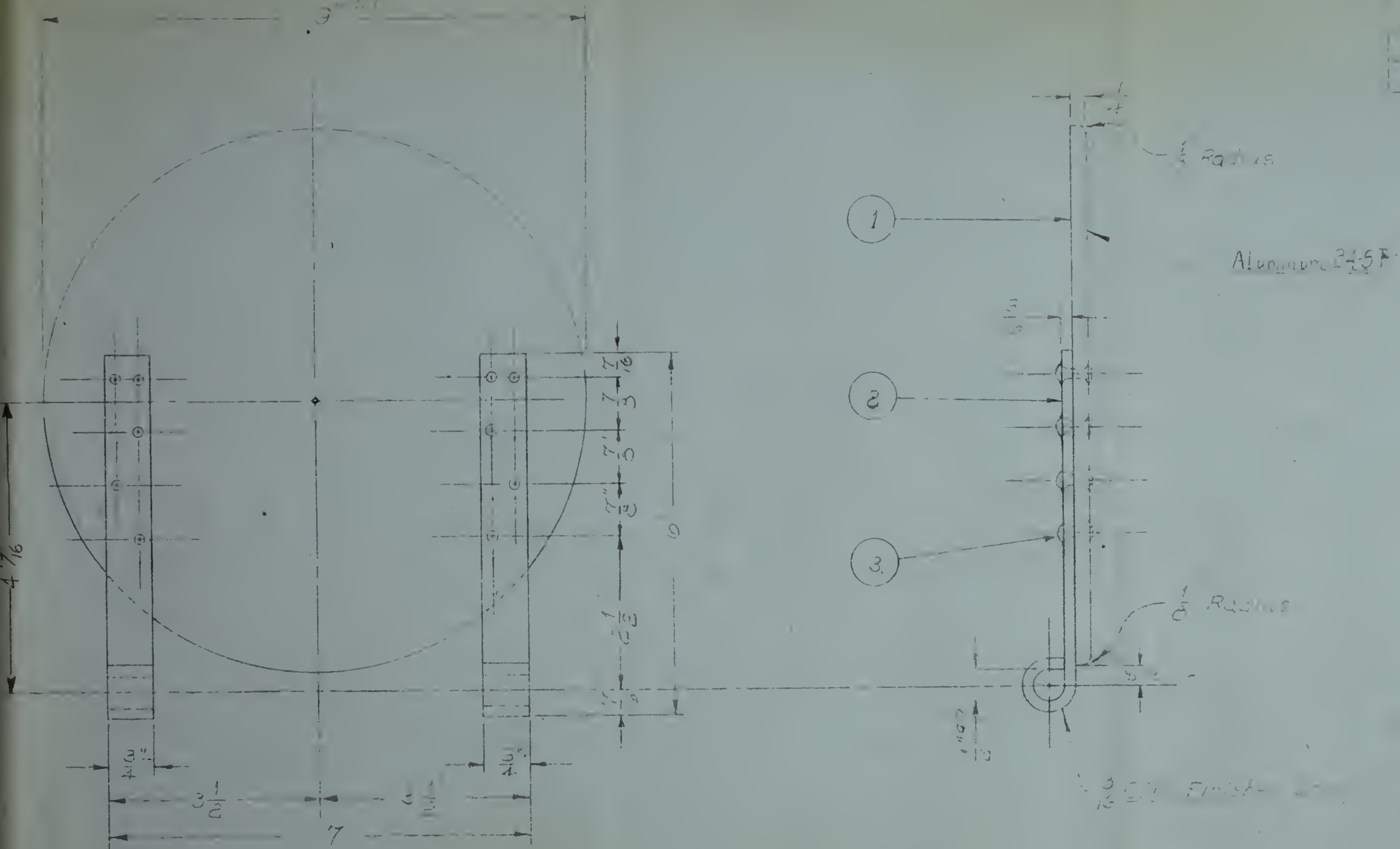
7
8
10
9
5°
6"
20



NO.	MATERIAL	NO REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
LOWER TUNNEL ADAPTER FLANGE ASSEMBLY		
DRAWN BY Busner	DATE 5/11/49	
CHECKED BY [illegible]	SCALE Half Size	
NEXT ASSEM. 0-1268	B 4260	

FIG. 4

1.	C 2261	EMERGEN[illegible] COVER	1
2.		HINGE	3
3.	3 [illegible]	BUTTON HD. RIVET	[illegible]



NO.	MATERIAL	NO REQ.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

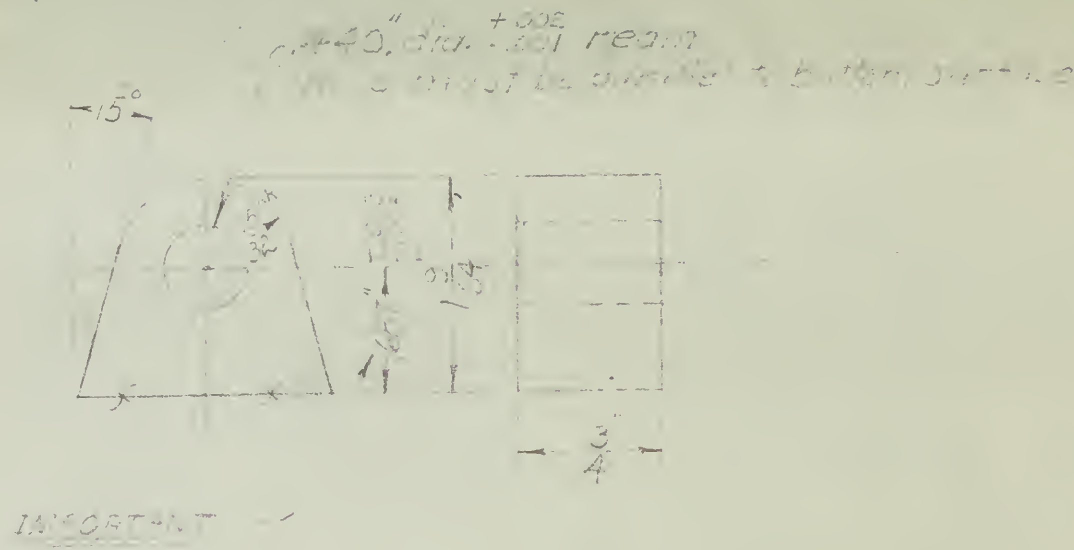
SAFFTY VALVE

DRAWN BY [illegible] | DATE [illegible]

CHECKED BY [illegible] | SCALE [illegible]

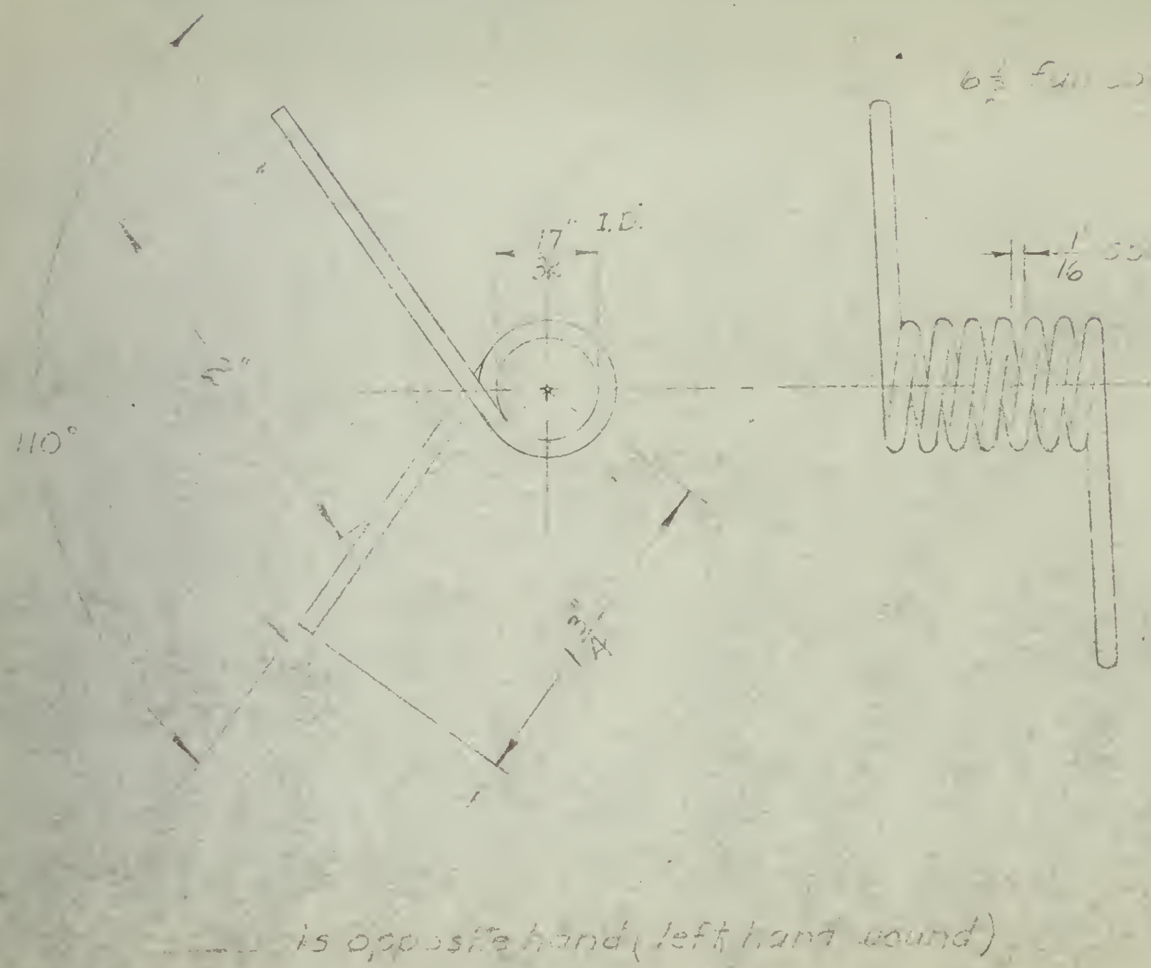
NEXT ASSEM. D 4[illegible]0 | D - 2261

FIG. 5



NO.	MATERIAL	NO. REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
BEARING BLOCK, SAFETY VALVE HINGE		
DRAWN BY [illegible]	DATE 5/10/49	
CHECKED BY [illegible]	SCALE	
NEXT ASSEM. C-4260	0 1262	

FIG. 6



NO.	MATERIAL	NO. REQ.
1	Same Spring [illegible] Hand Wound	1
2	.093 dia. Hard Drawn Spring Steel Wire Right Hand	1

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

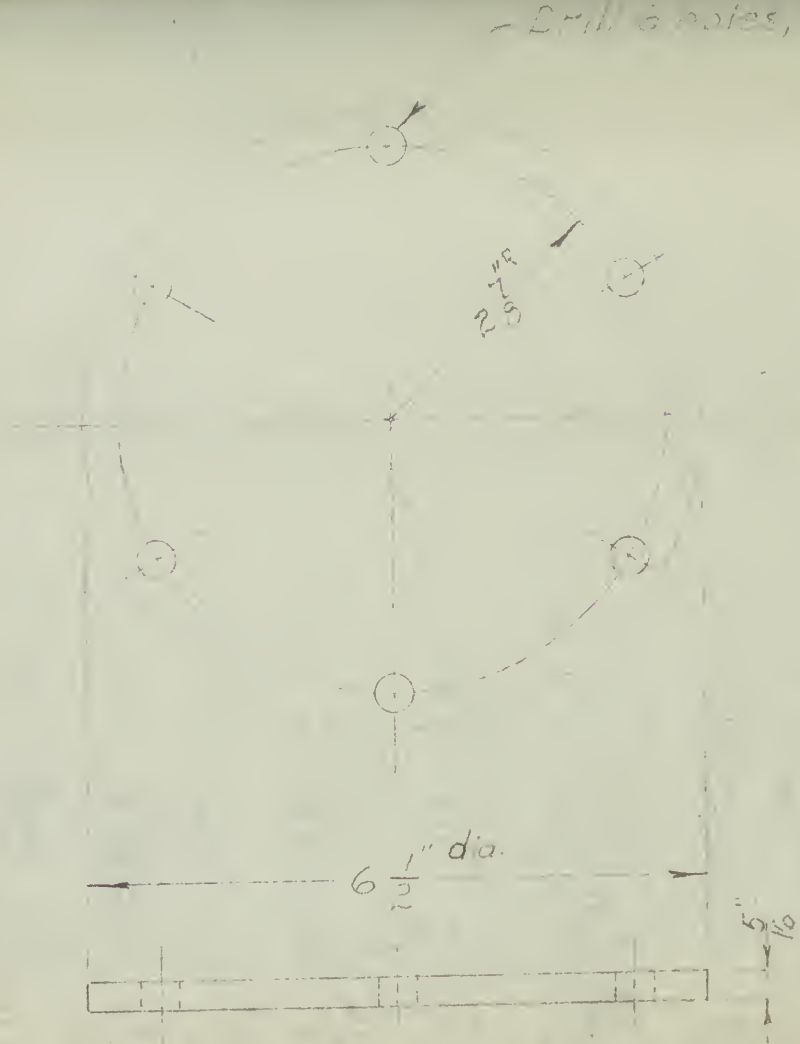
SAFETY VALVE SPRINGS

DRAWN BY Burns	DATE 5/10/49
CHECKED BY [illegible]	SCALE Full
NEXT ASSEM. O-4260	O-1263

FIG. 7

NO.	MATERIAL	NO. REQ.
		1
MASSACHUSETTS INSTITUTE OF TECHNOLOGY GAS TURBINE LABORATORY		
SAFETY VALVE [illegible] P[illegible]		
DRAWN BY [illegible]	DATE 5/10/49	
CHECKED BY [illegible]	SCALE FULL	
NEXT ASSEM. 0-4265	0-1264	

FIG. 8



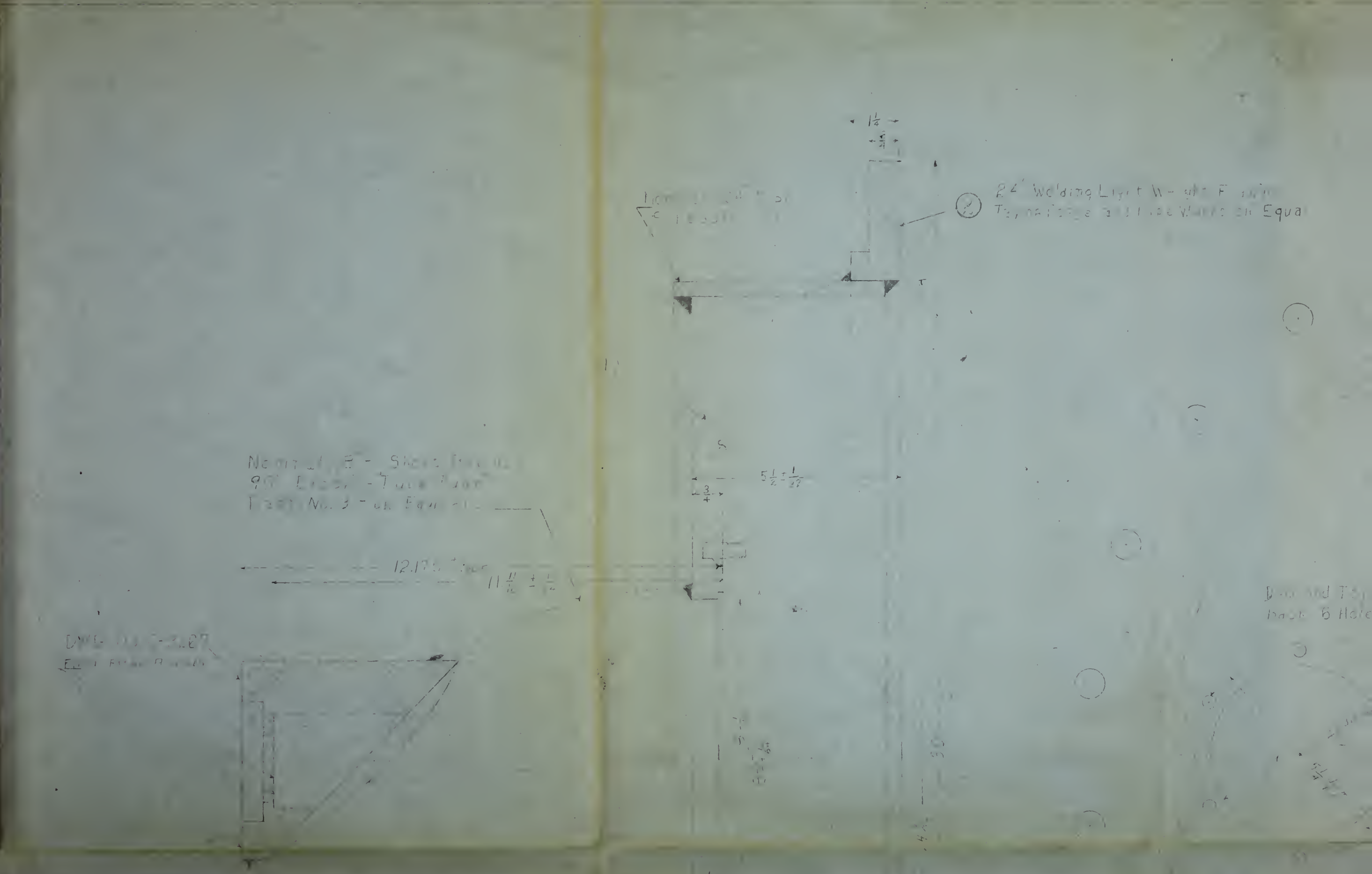
	5/8 ...	1
NO.	MATERIAL	NO. REQ

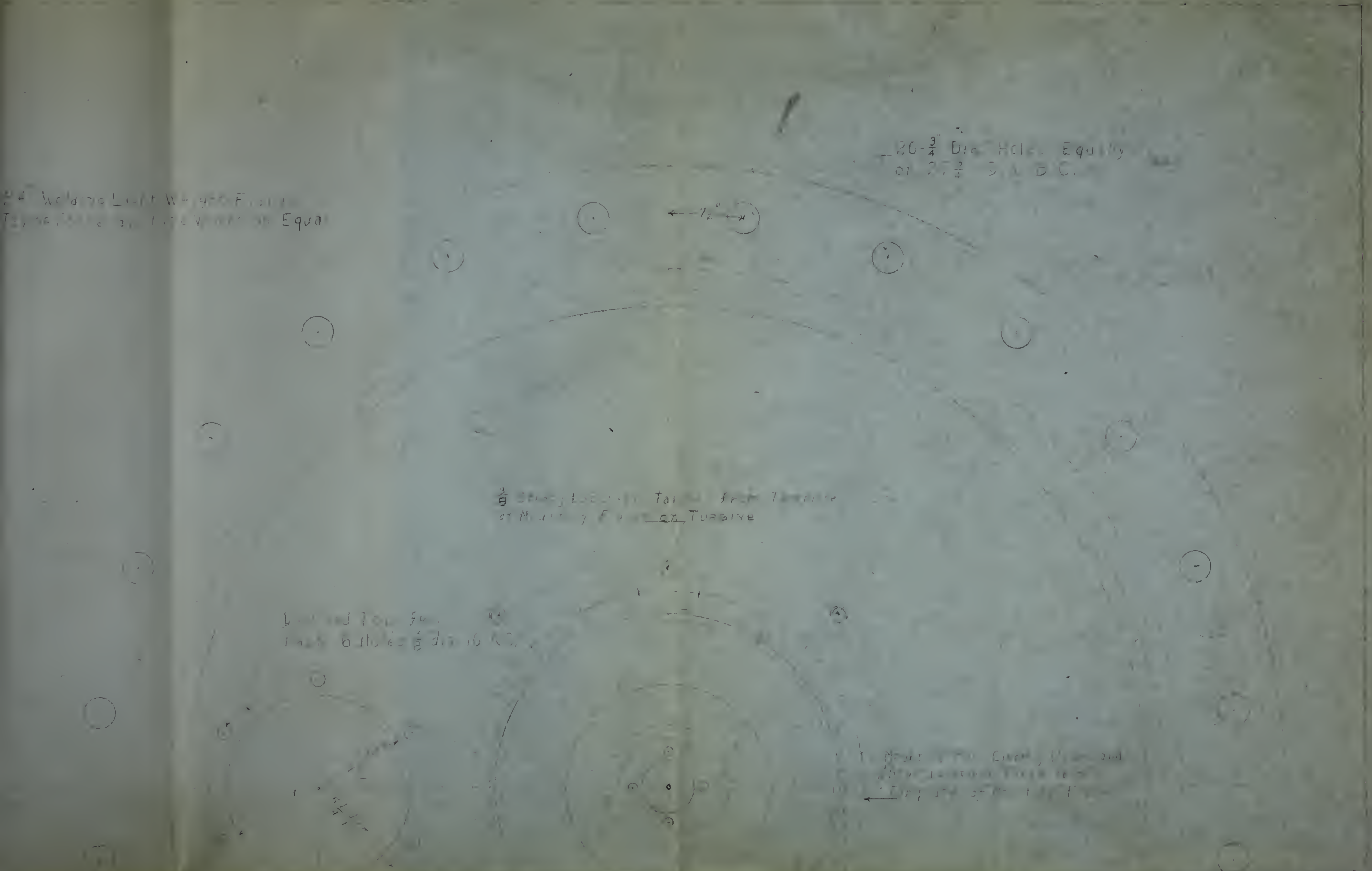
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY

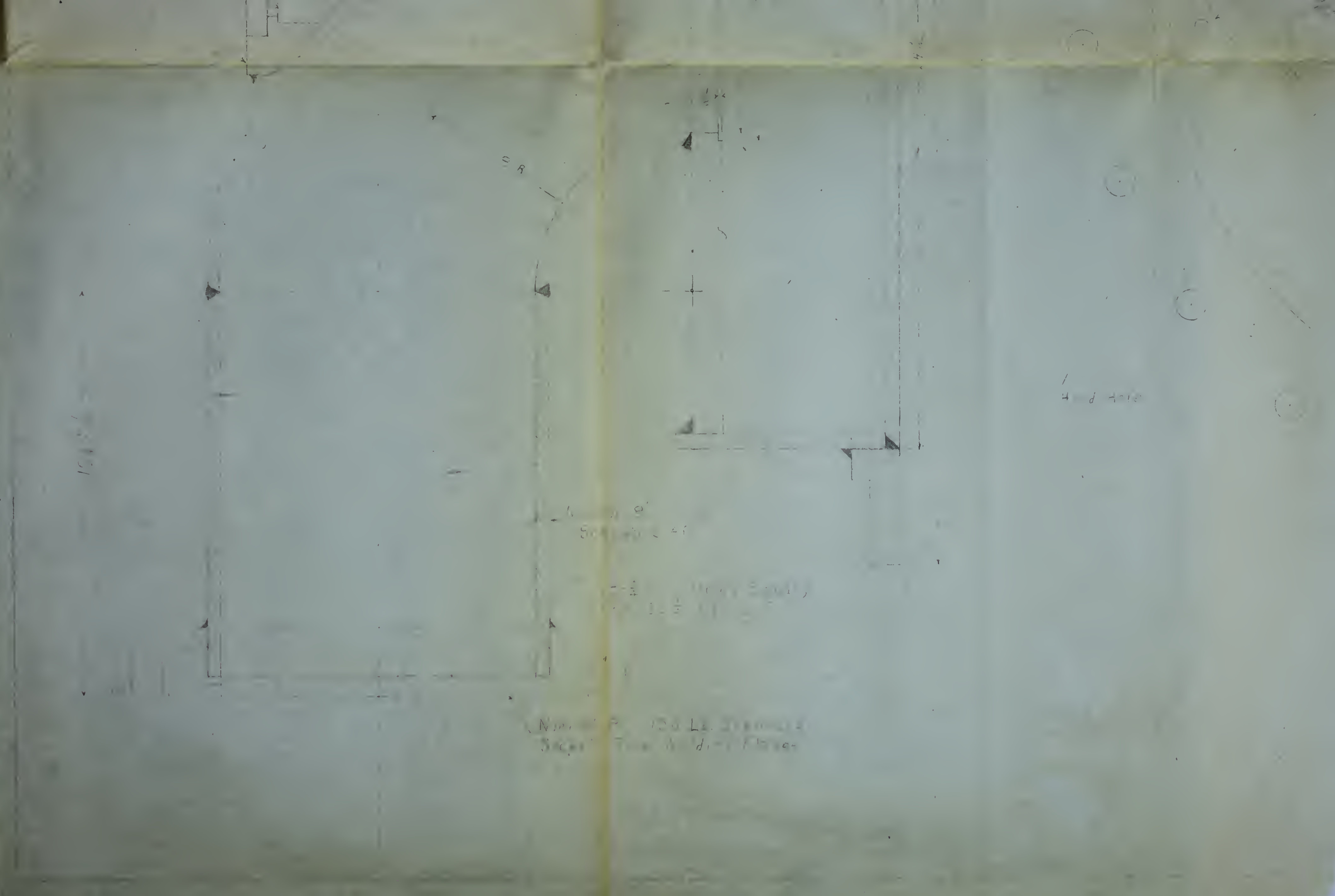
HAND HOLE COVER PLATE,
Lower Tunnel Adapter

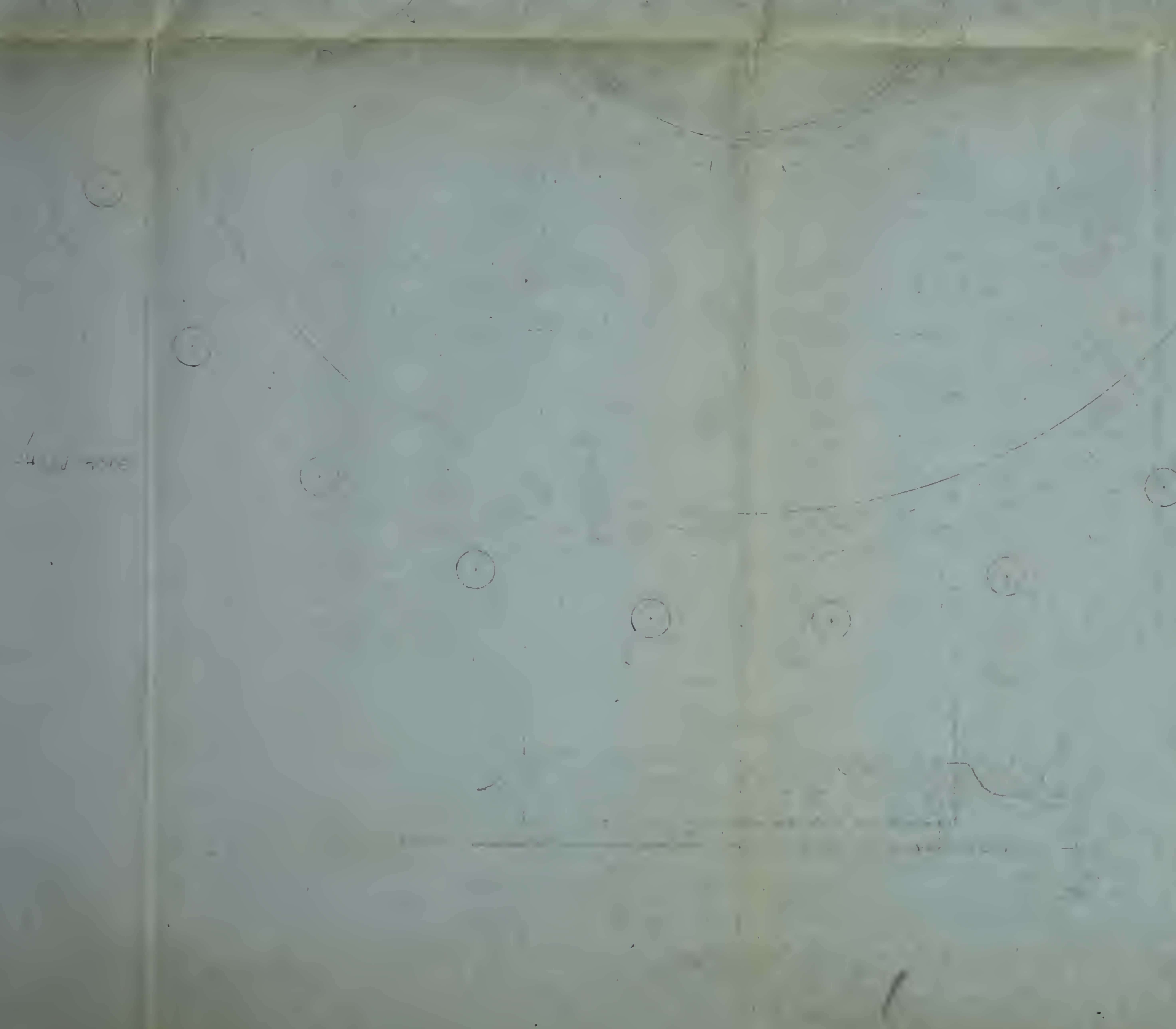
DRAWN BY Burns	DATE 5/10/49
CHECKED BY [illegible]	SCALE HALF SIZE
NEXT ASSEM. 0-4260	0-1265

FIG. 9




20-¾" Dia. Holes Equally
Equal
Turbine





NO.	MATERIAL	NO. REQ.
	Cold Finished Steel	
1	Turbine Mounting Flange	1
2	24' [illegible]	2

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GAS TURBINE LABORATORY

TUNNEL PIPING ASSEMBLY

DRAWN BY	[illegible]	DATE	[illegible]
CHECKED BY	[illegible]	SCALE	[illegible]
NEXT ASSEM	D-4260 D-5659	D 4266	

FIG. 10

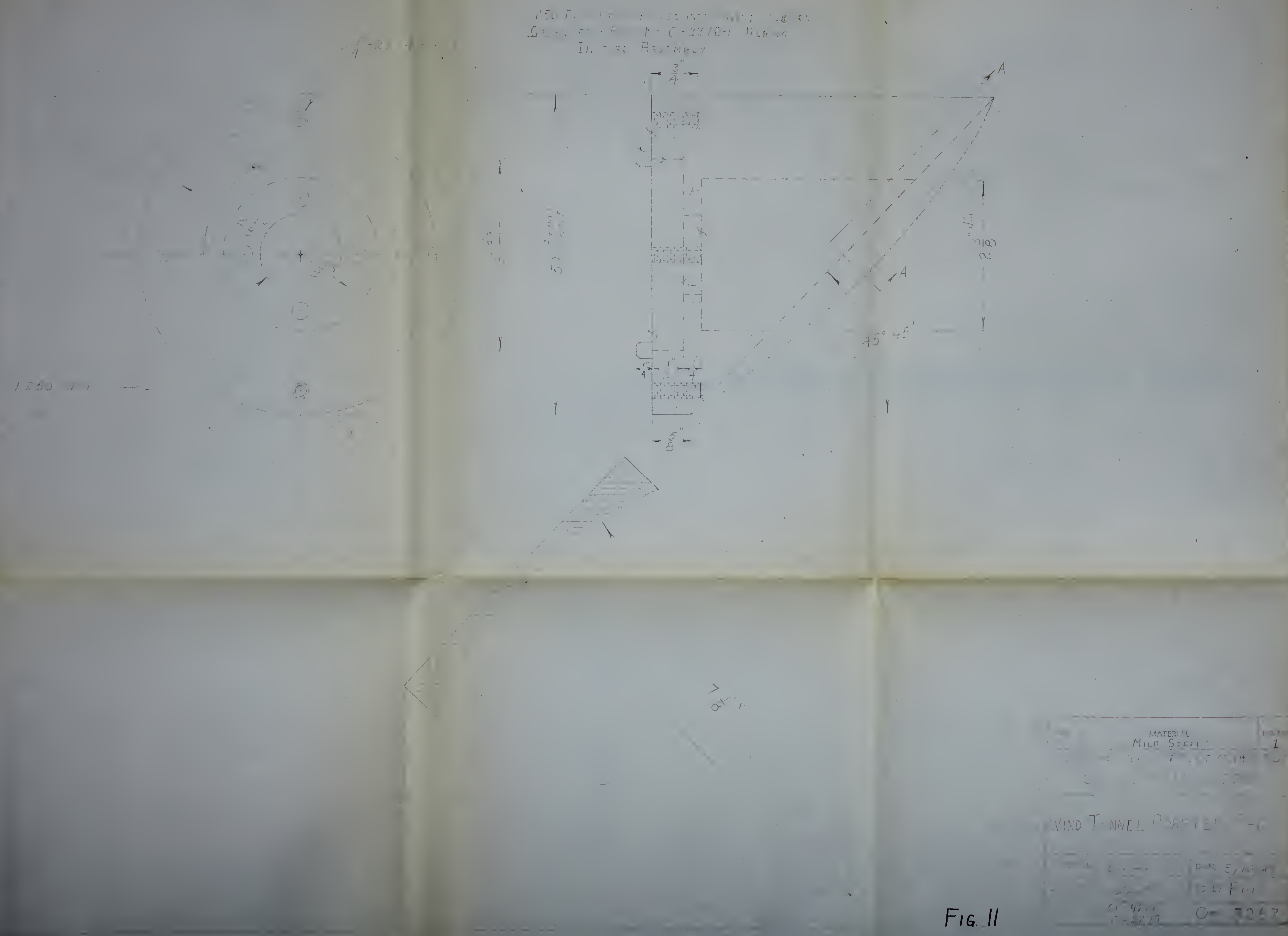


FIG. 11

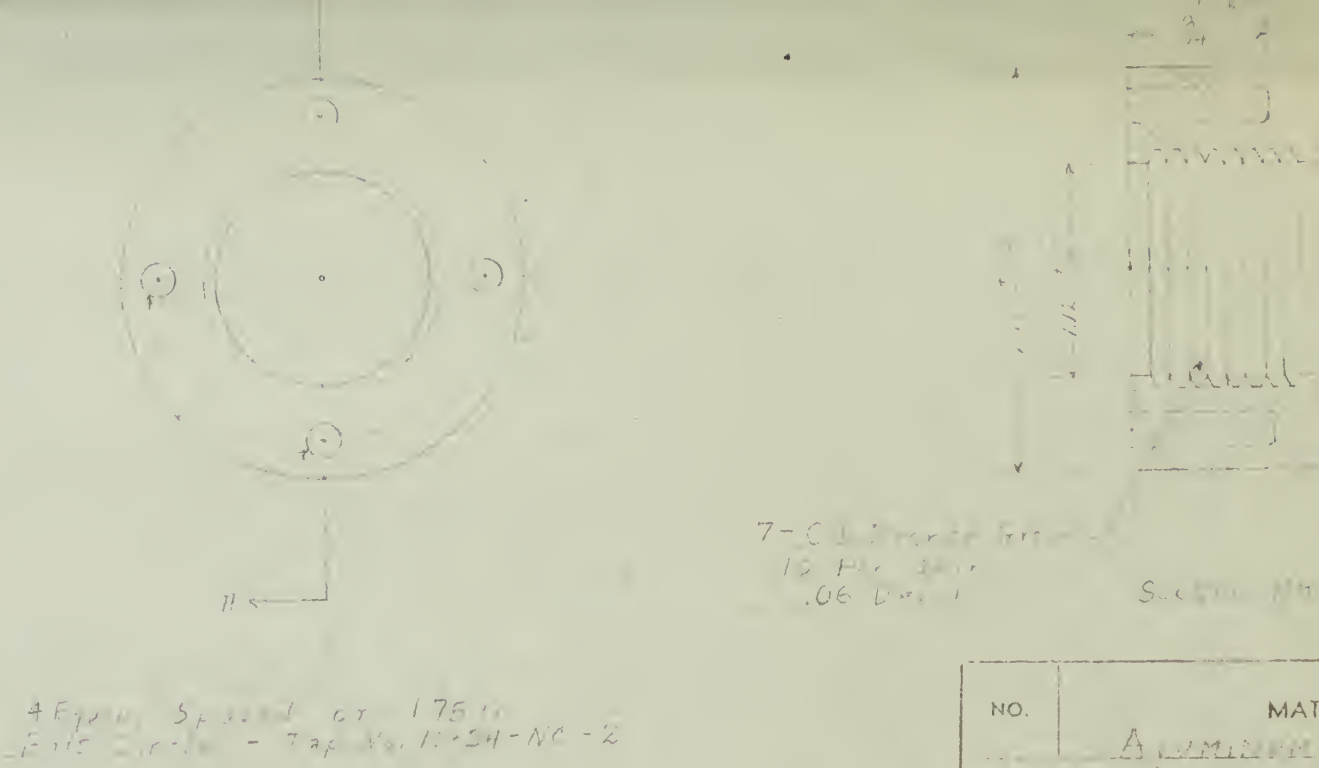
NO.	MATERIAL	NO REQ
		1

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

DRAWN BY	DATE
CHECKED BY	SCALE
NEXT ASSEM.	O- 1268

FIG. 12



NO.	MATERIAL	NO. REQ.
	Aluminum Bronze	1

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

FORWARD SHAFT EXTENSION SEAL

DRAWN BY	DATE 5/ /47
CHECKED BY	SCALE FULL
NEXT ASSEM. 0-5259	0F 1269

FIG. 13

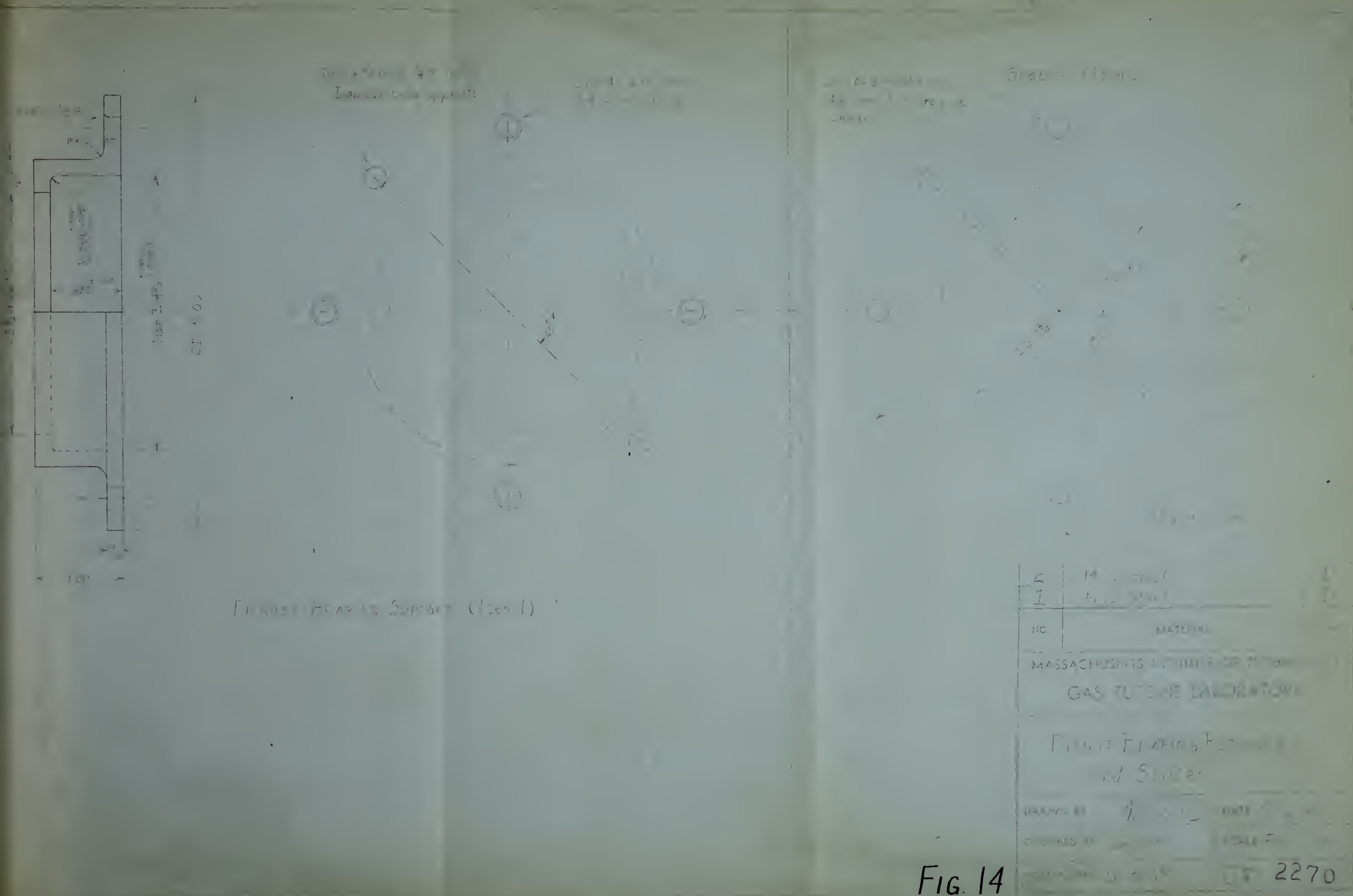
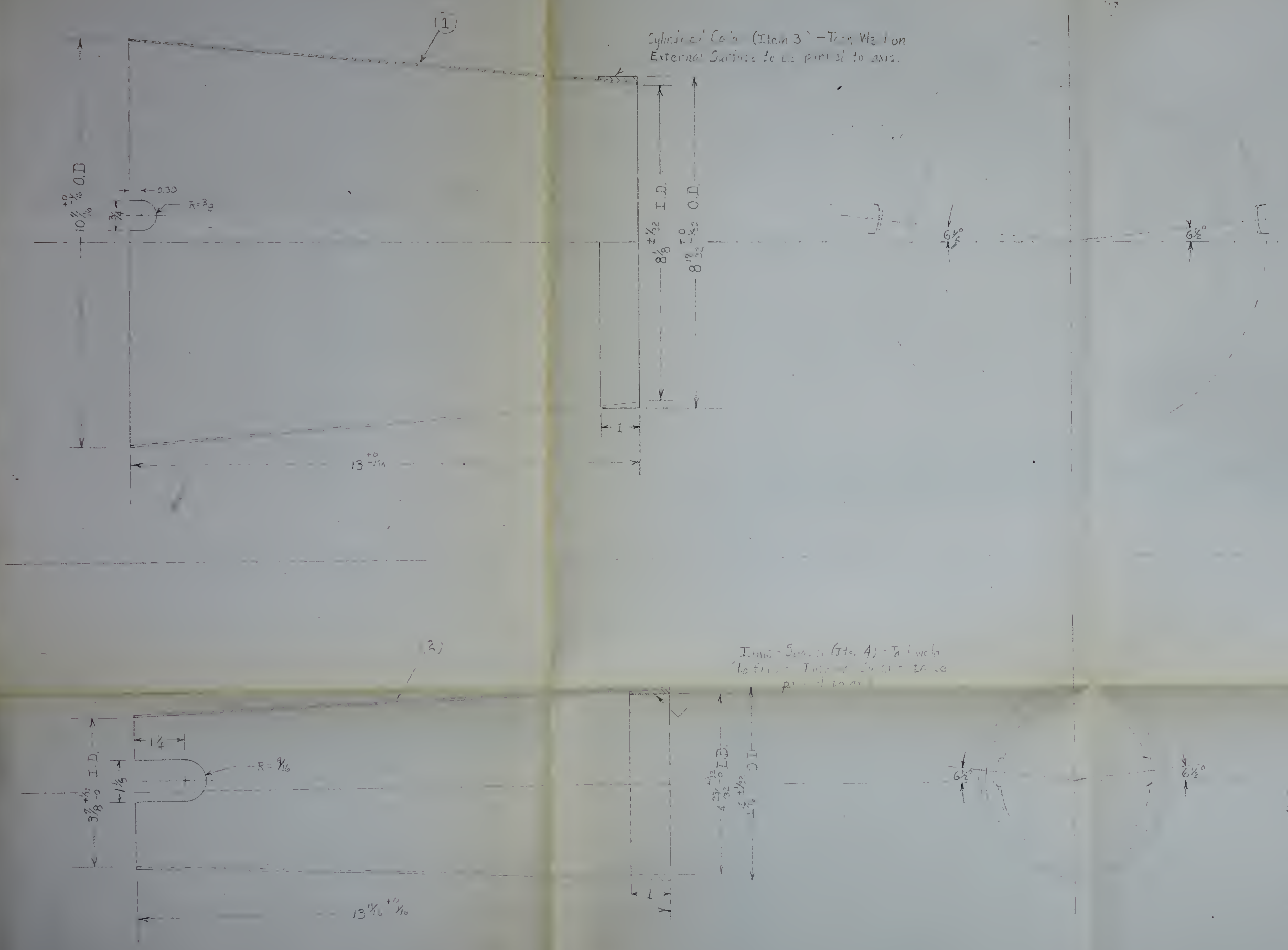


Fig. 14



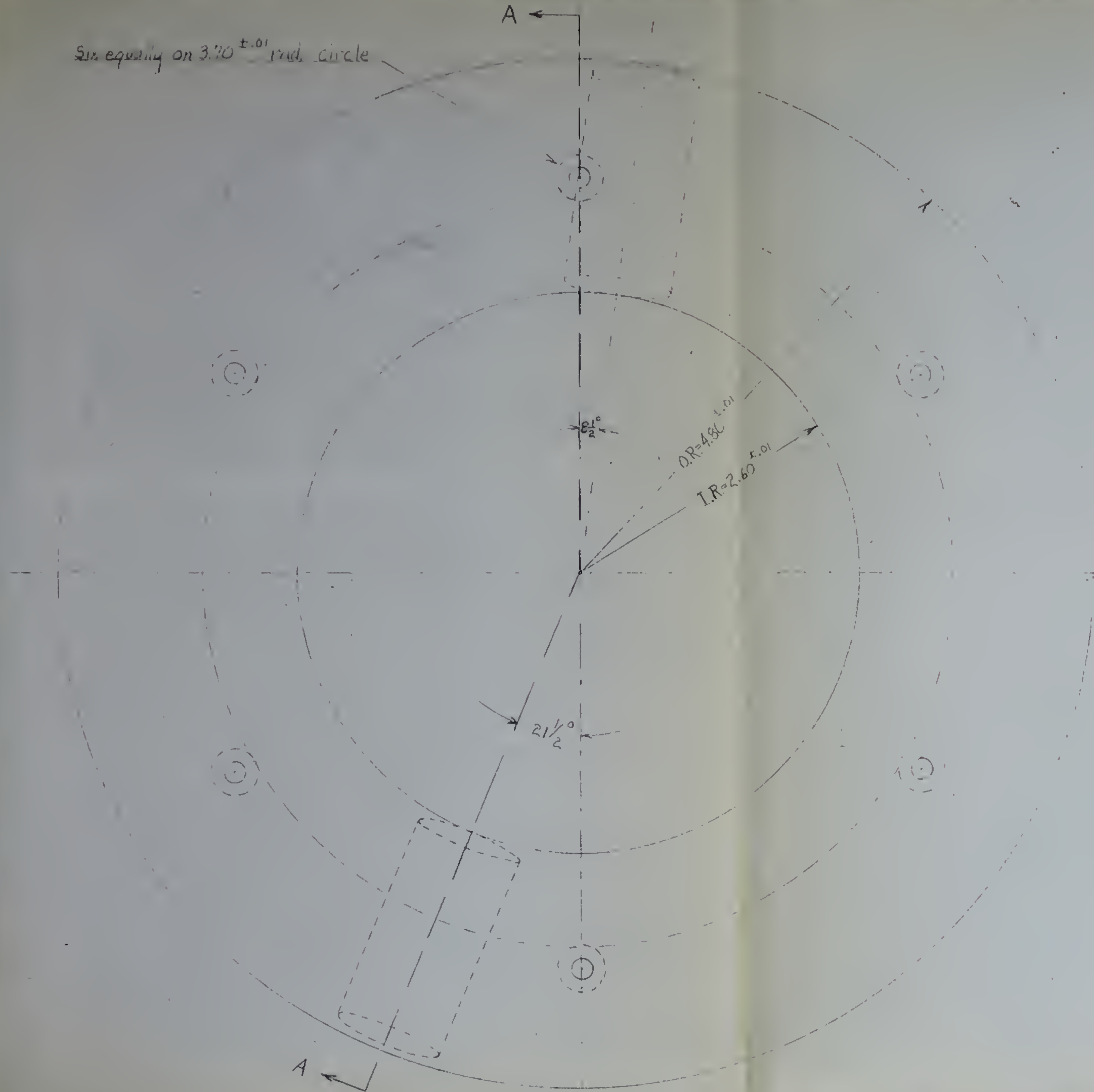
NO	MATERIAL	NO. REQ.
4	Mild Steel	1
3	Mild Steel	1
2	Sheet Metal – 19 gauge	1
1	Sheet Metal – 19 gauge	1

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY

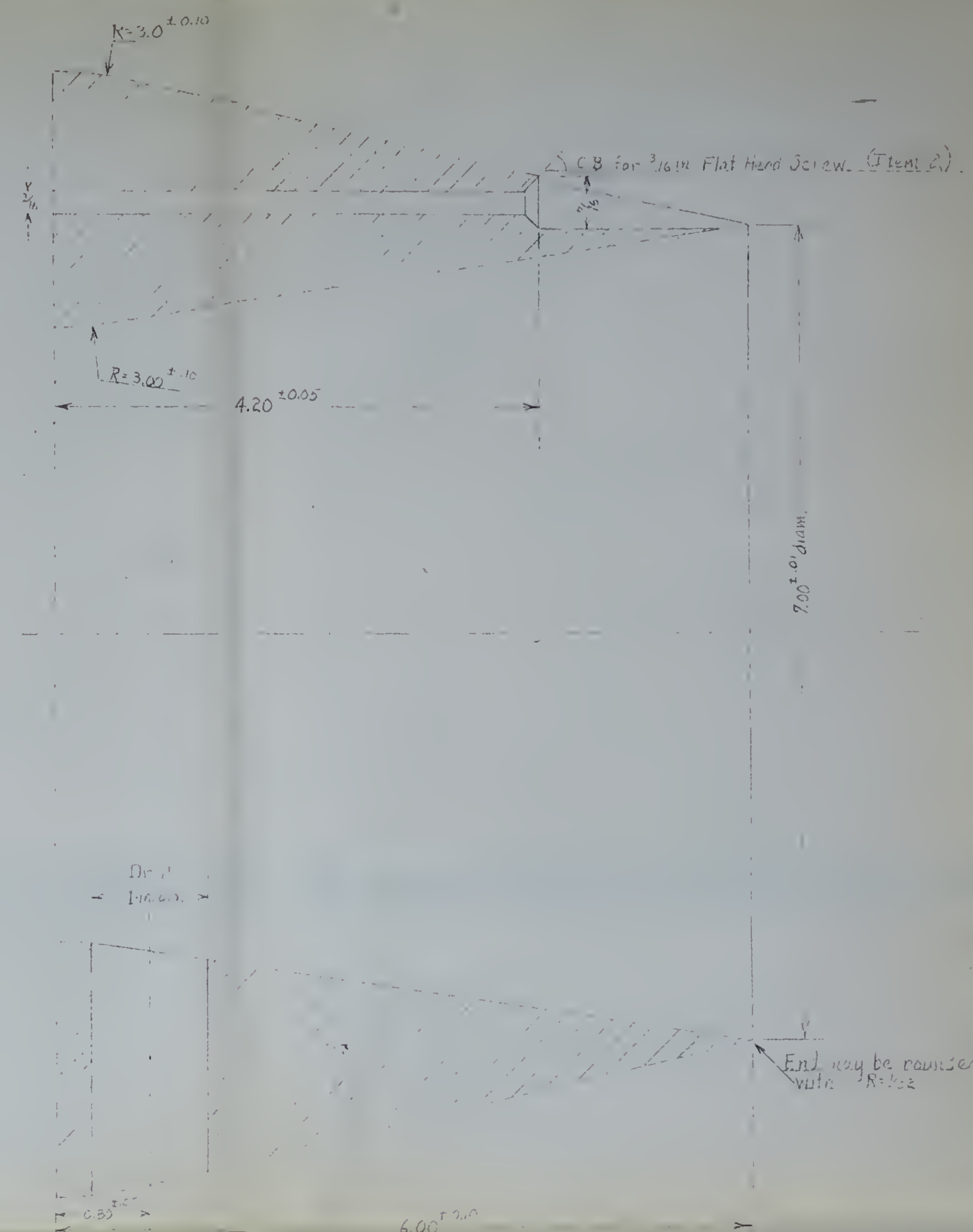
INNER FAIRINGS

DRAWN BY [illegible] DATE 5/10/49
CHECKED BY [illegible] SCALE Half Size
ASSEM 0-5259 0-2271

FIG. 15



Front End View
(Item 1)



Section A-A

Mat'l: Wood or Any Metal, as desired.

NO.	MATERIAL	NO. REQ.
2	4½ x 3/16 Flat Hd Screw (NF Thd)	6
1	Wood or Any Metal	1

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY

TAIL CONE

DRAWN BY Mousen DATE 5/10/47
CHECKED BY SCALE Full Size
NEXT ASSEM. D- 2272

FIG. 16

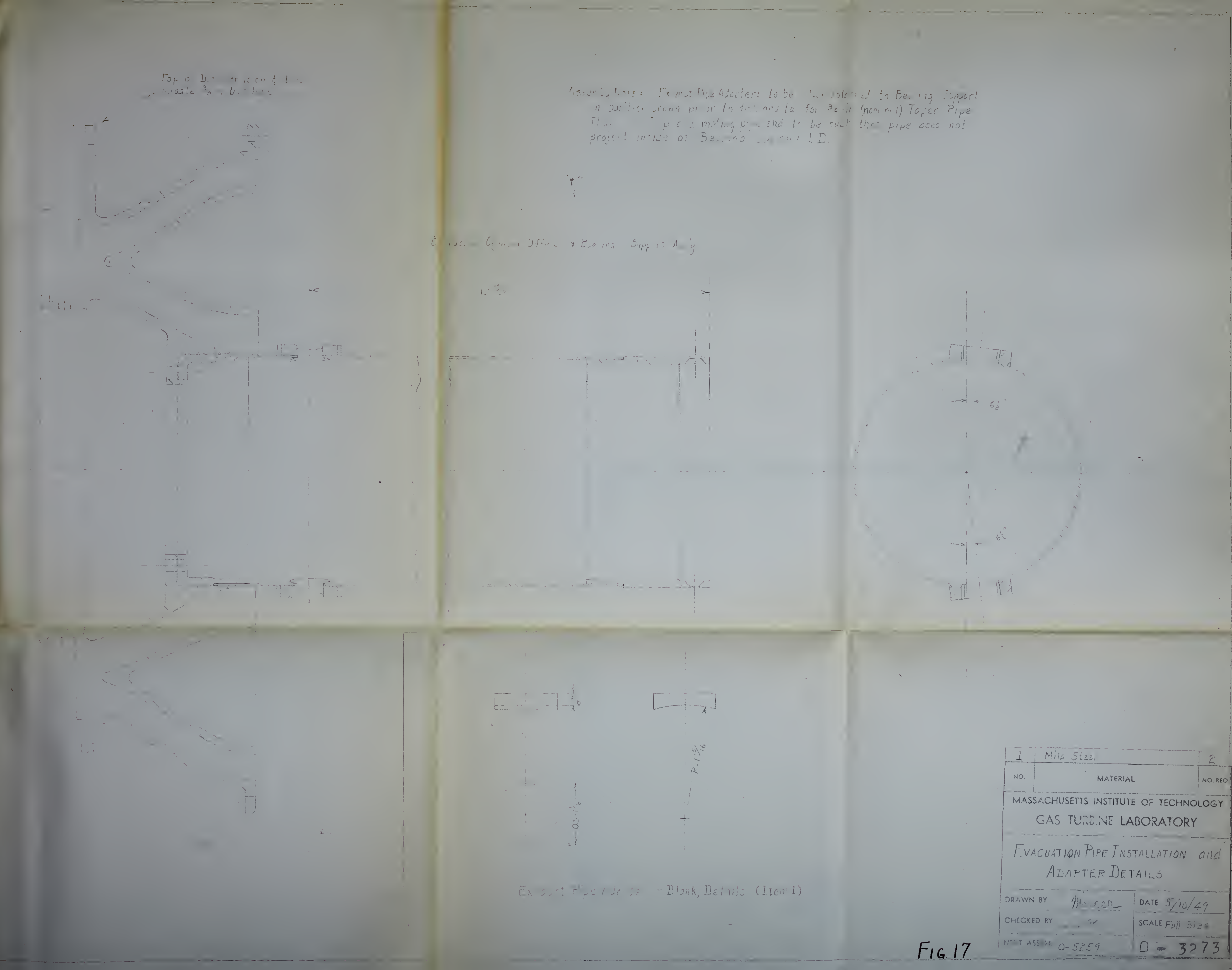
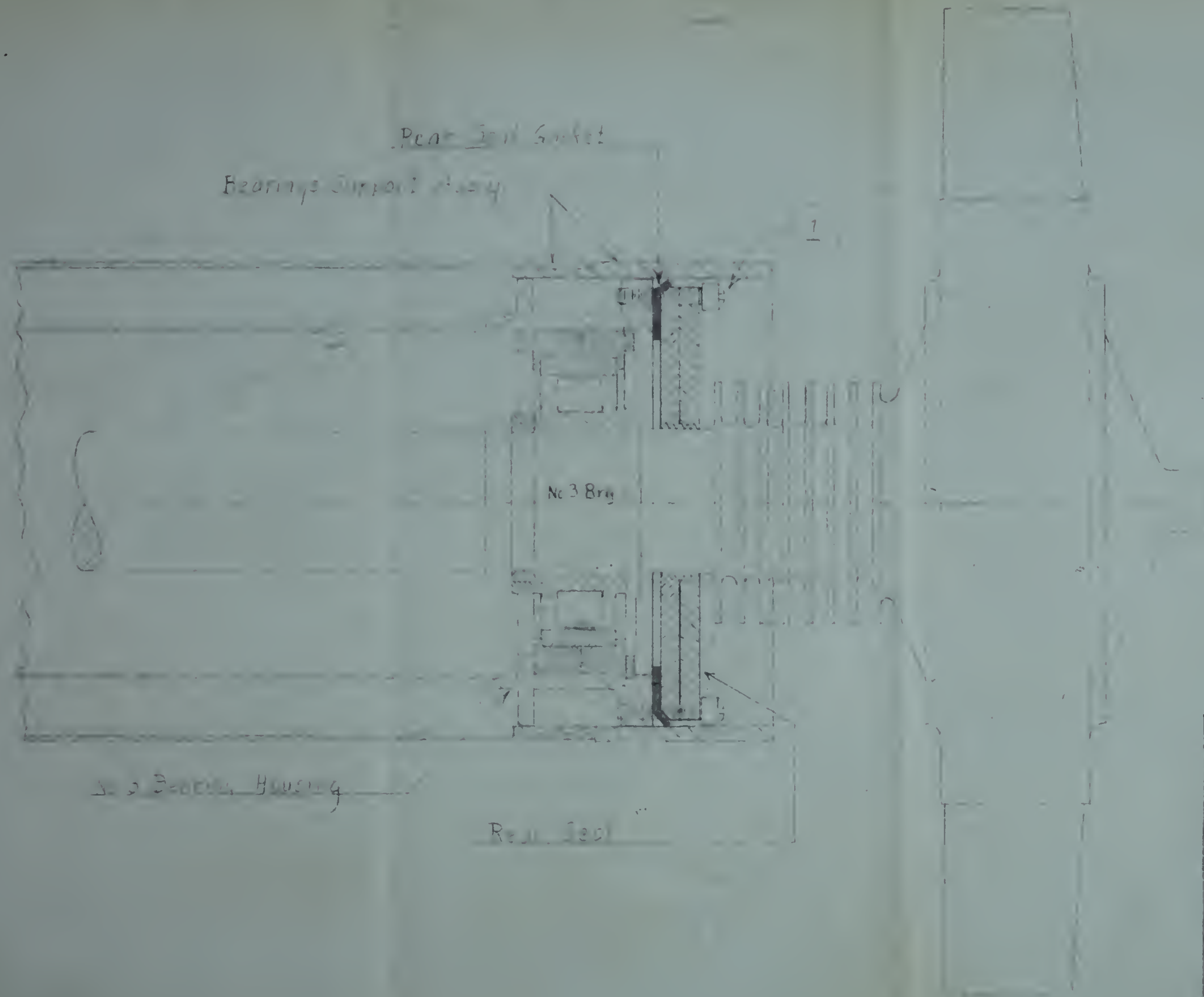

1
Mild Steel
2
NO.
MATERIAL
NO. REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY
EVACUATION PIPE INSTALLATION and ADAPTER DETAILS
DRAWN BY
DATE 5/10/49
CHECKED BY
SCALE Full Size
NEXT ASSEM. 0-5259
D - 3273

FIG. 17



1	Filister Hd Cap Screw, 1in ...	4
NO.	MATERIAL	NO. REQ

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GAS TURBINE LABORATORY

REAR SEALS - SUB ASSEMBLY

DRAWN BY	DATE 5/16/49
CHECKED BY	SCALE Full Size
NEXT ASSEM. C-2255	D - 2274

Fig. 18

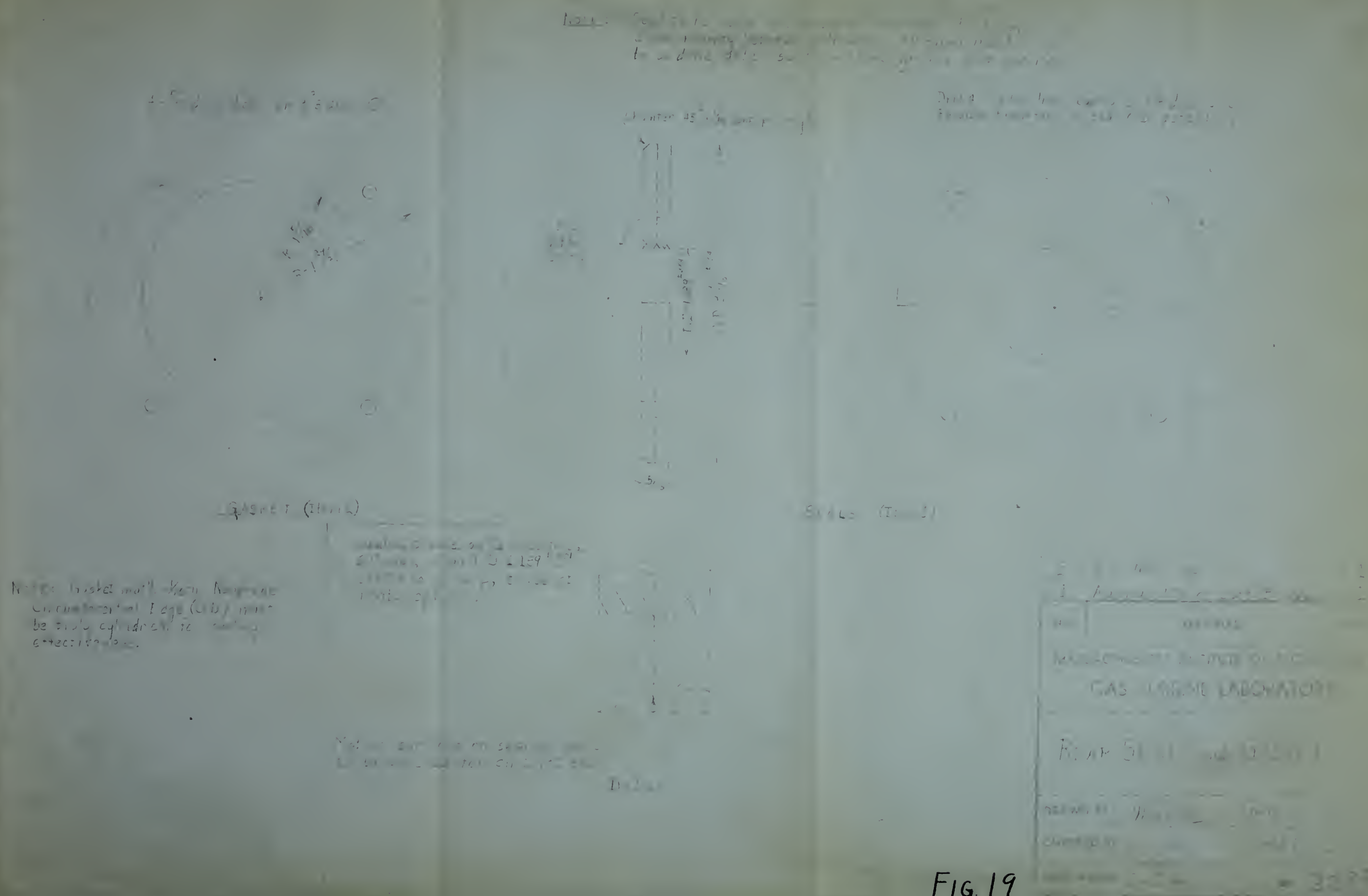
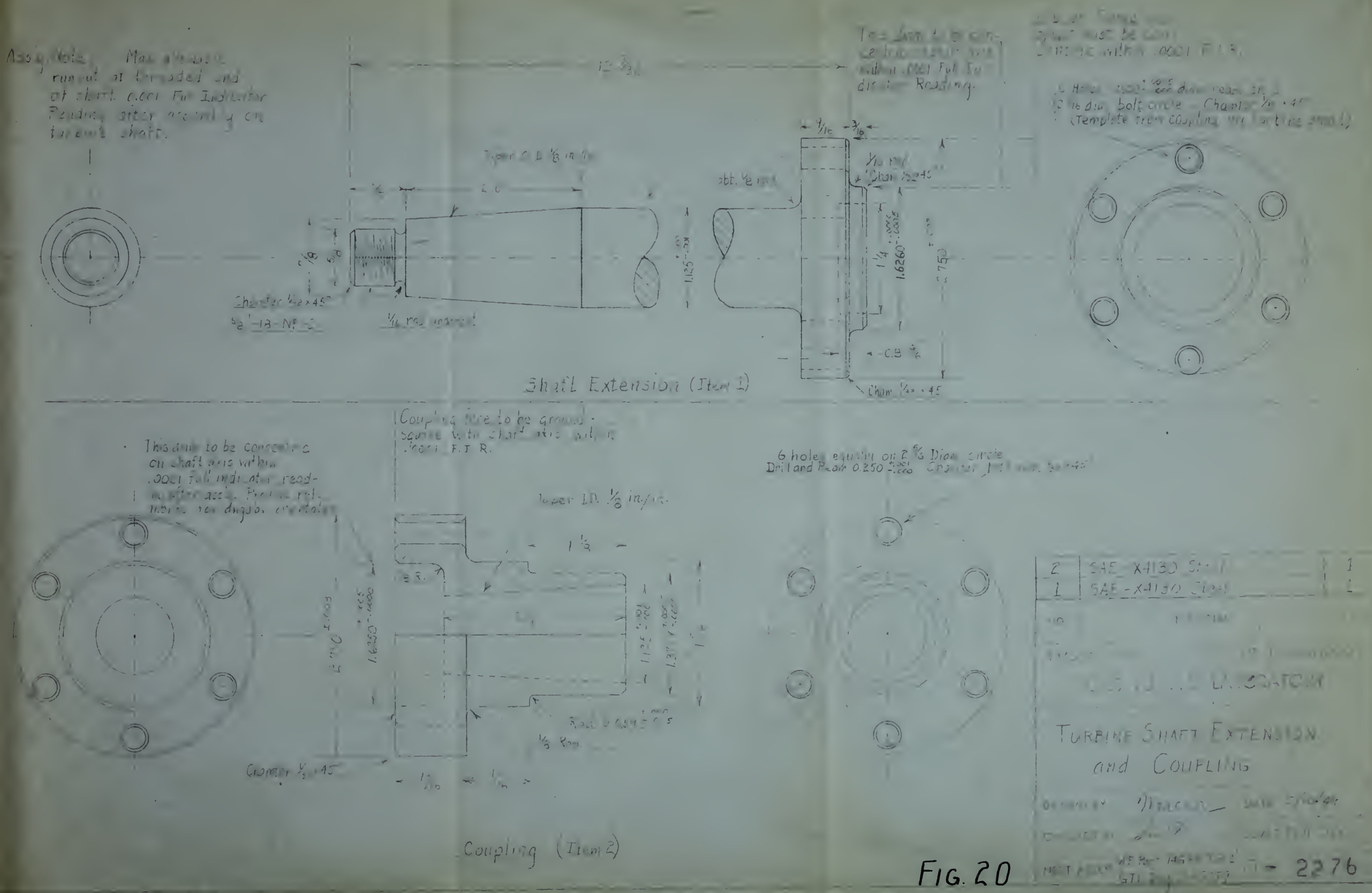
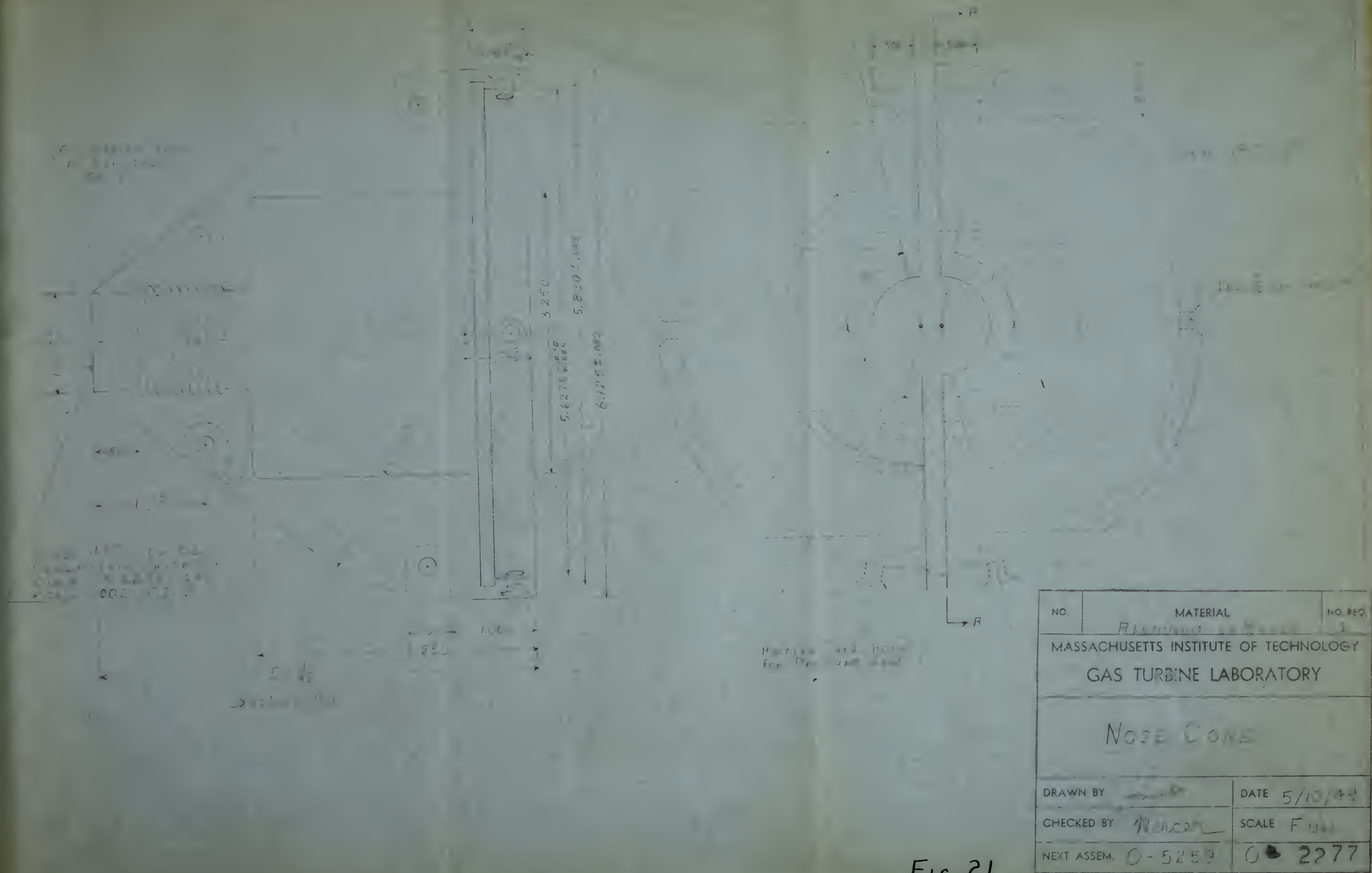


FIG. 19



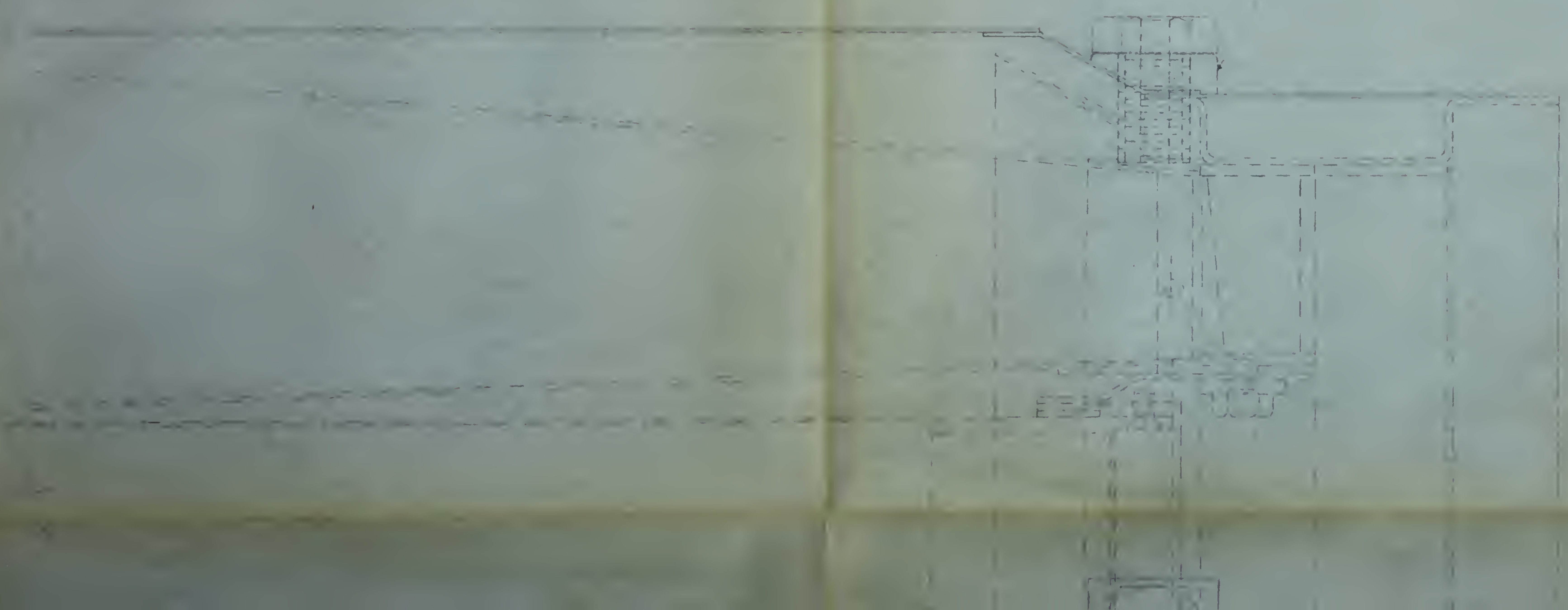
Shaft Extension (Item 1)
Coupling (Item 2)
Taper I.D. 1/8 in./in.
6 holes equally on 2 9/16 Dia. circle
1.6260
1.125
1 1/4
2.750
Chamfer 1/32 x 45°
TURBINE SHAFT EXTENSION
and COUPLING
SAE - X4130
2276

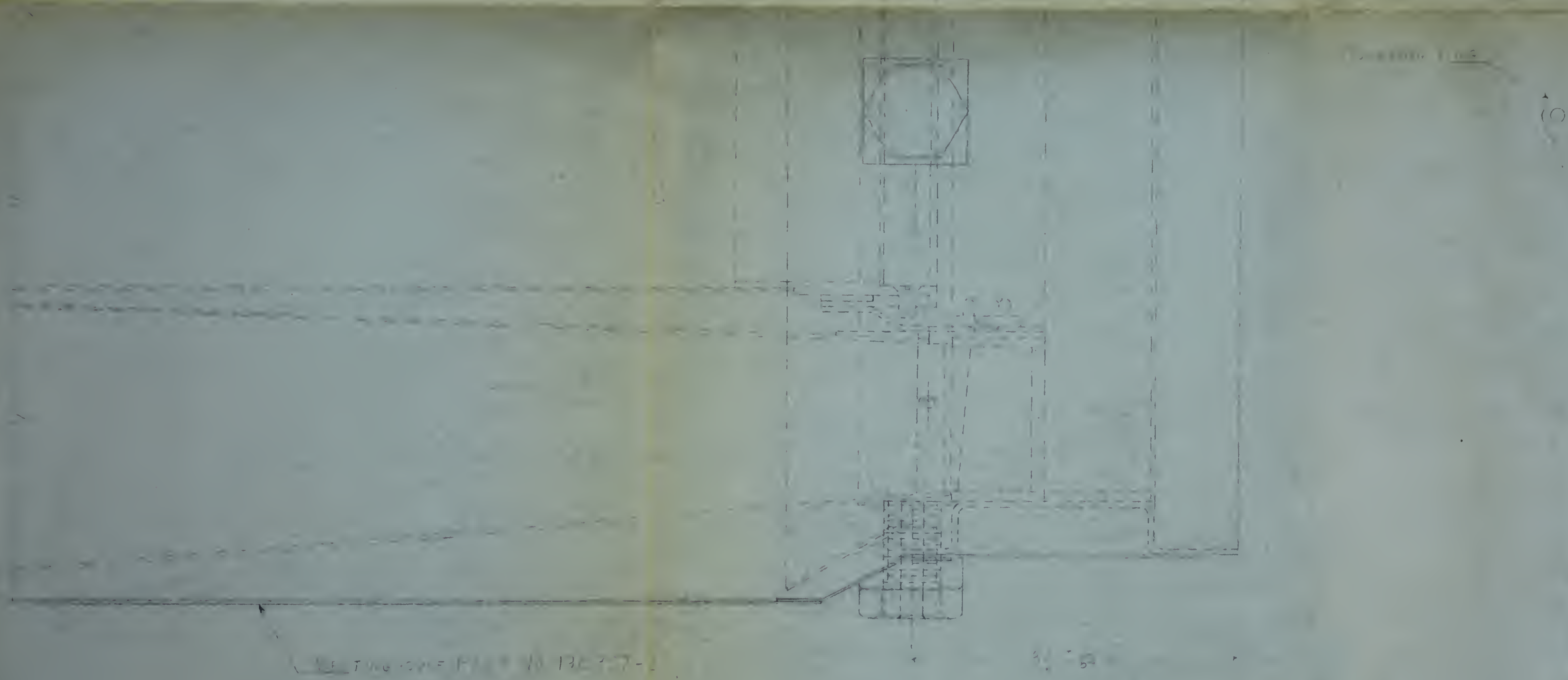
FIG. 20



NO.
MATERIAL
NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY
NOSE CONE
DRAWN BY
DATE
CHECKED BY
SCALE
NEXT ASSEM.
2277
3.250

FIG. 21





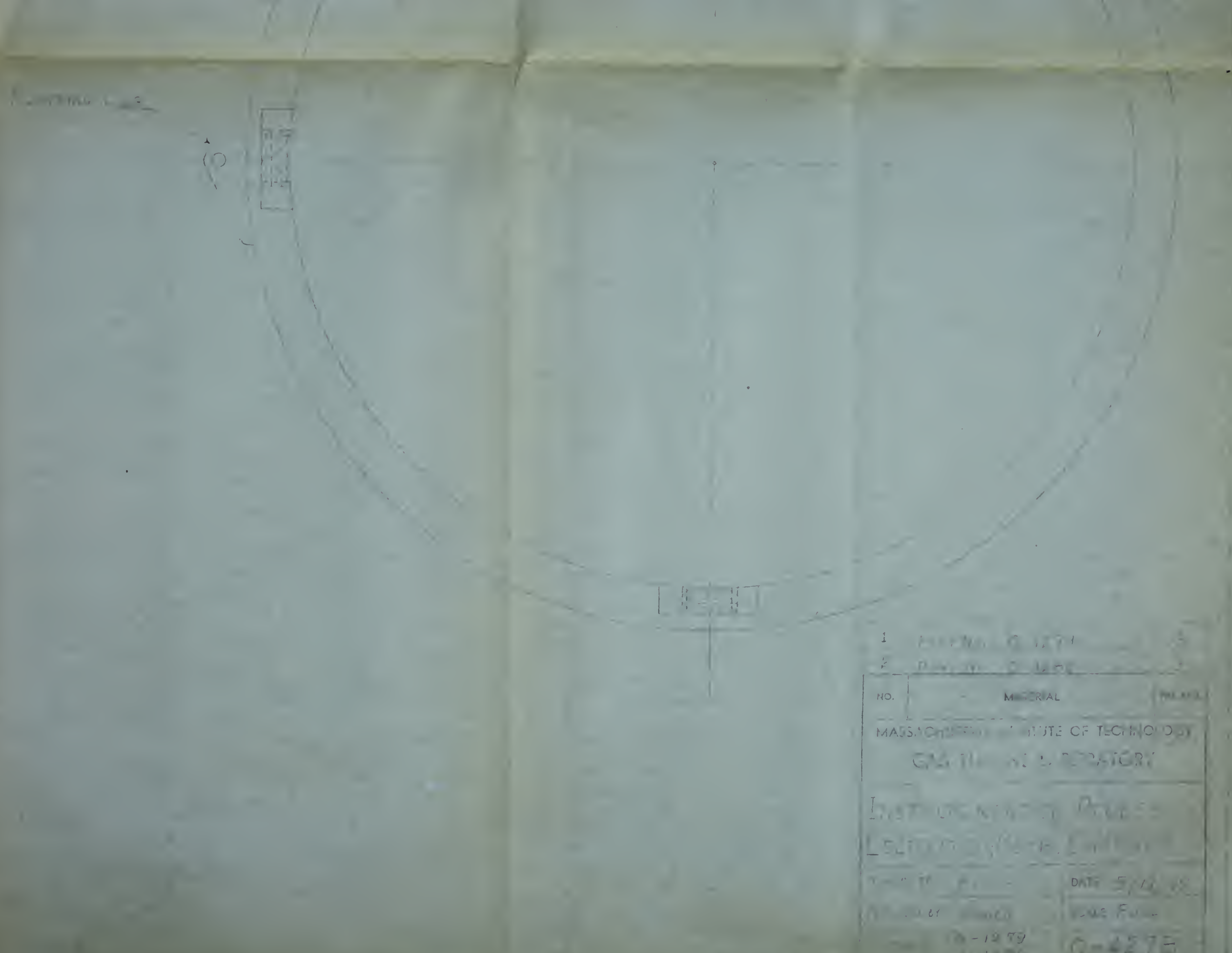


FIG. 22

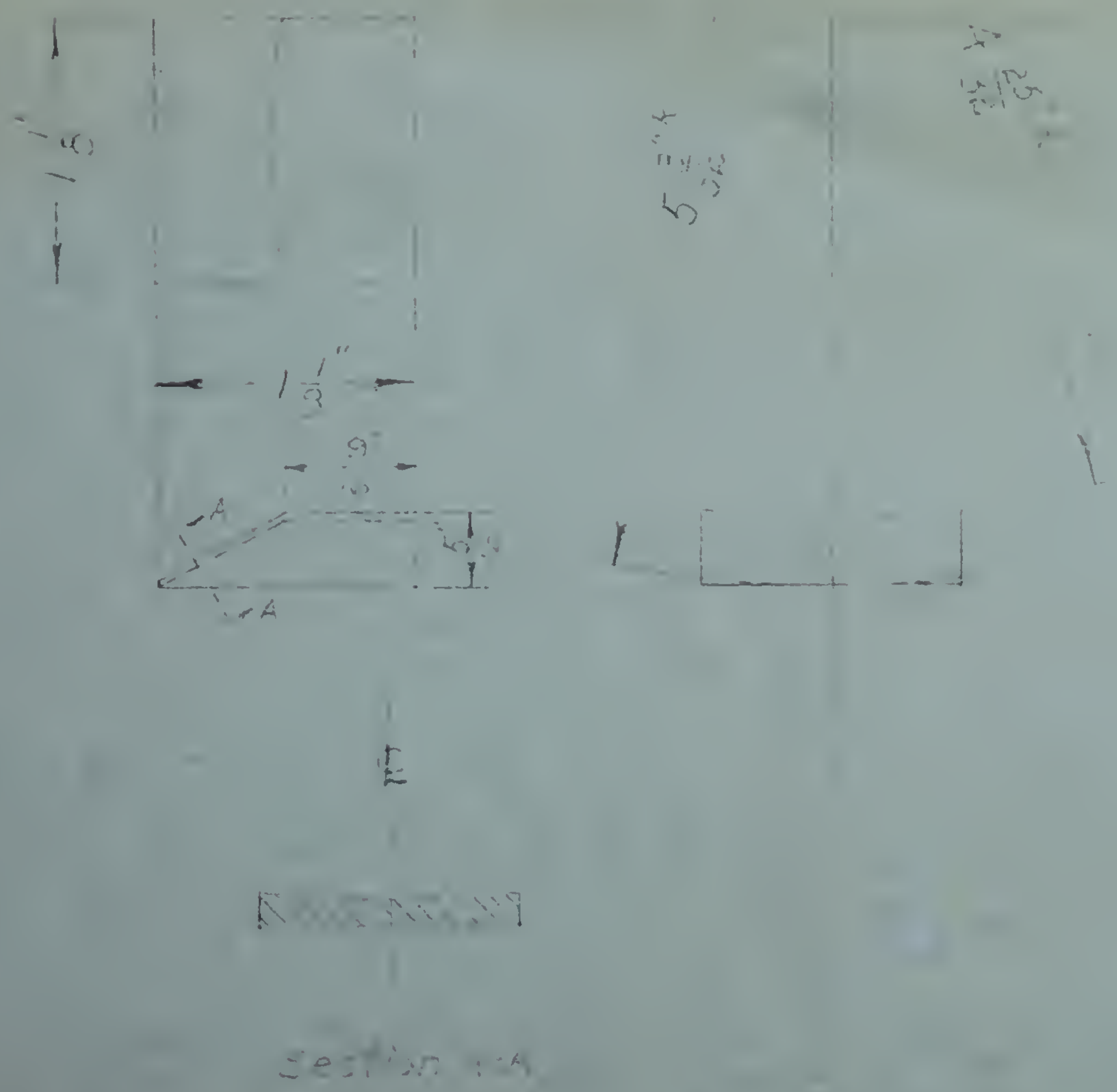


Fig. 24

1	Mild Steel	3
NO.	MATERIAL	NO. REQ.

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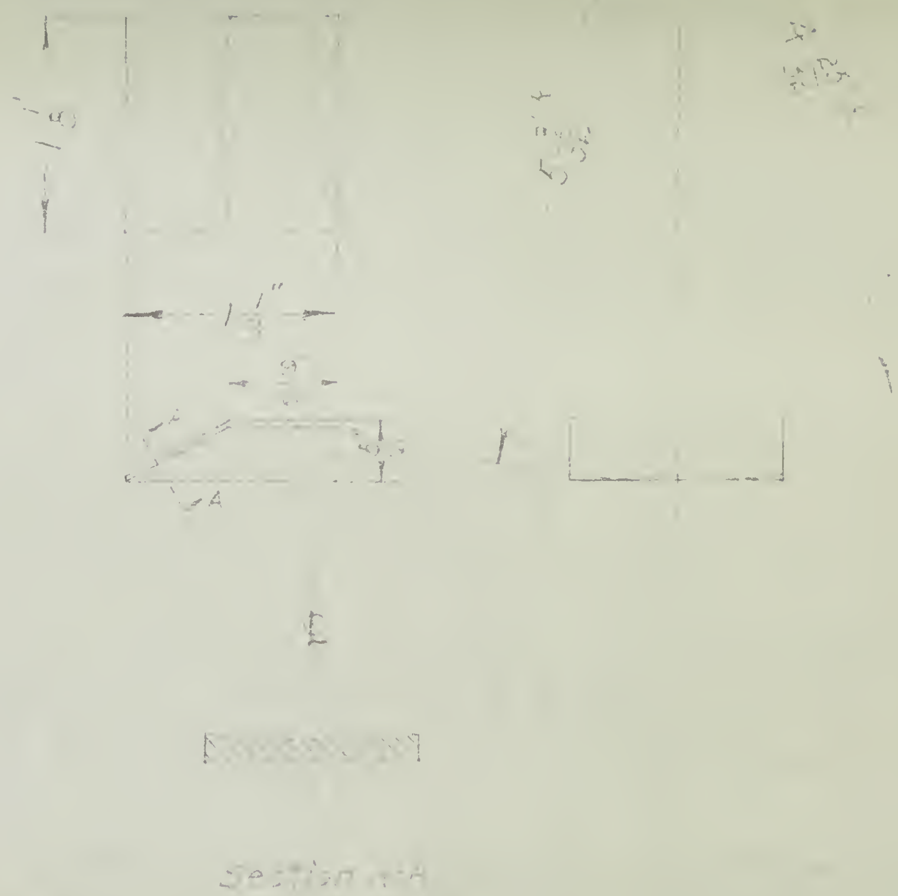
GAS TURBINE LABORATORY

INSTRUMENT ACCESS PAD, BLANK.
COMBUSTION CHAMBER

DRAWN BY Burns	DATE 5/11/49
CHECKED BY Mercer	SCALE Full Size
NEXT ASSEM. D-4278	D-1280



1	Mild Steel	3
NO.	MATERIAL	NO. REQ

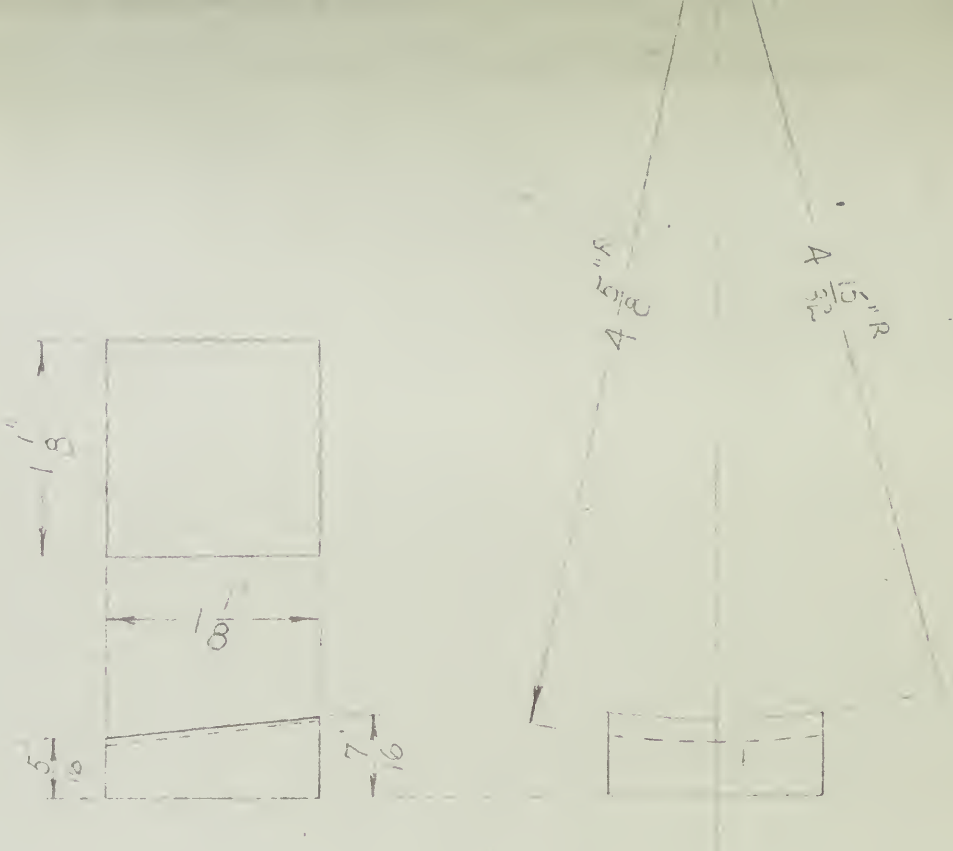
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

INSTRUMENT ACCESS PAD, BLANK,
COMBUSTION CHAMBER

DRAWN BY Burns	DATE 5/11/49
CHECKED BY Mercer	SCALE Full Size
NEXT ASSEM. D-4278	D- 1280

FIG. 24



NO. | MATERIAL | 4

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

GAS TURBINE LABORATORY

PAD,

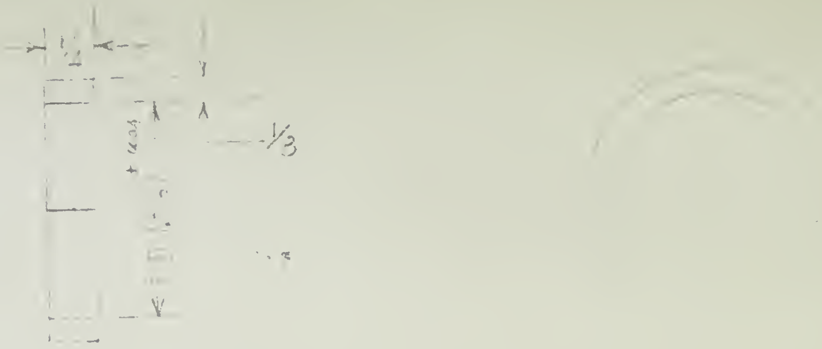
TAIL PIPE

DRAWN BY | DATE 5/10/49

CHECKED BY | SCALE FULL

NEXT ASSEM O-3262

Fig. 25



To be shrunk onto Turbine [illegible]
Extension [illegible]
after [illegible] Thrust Bearing

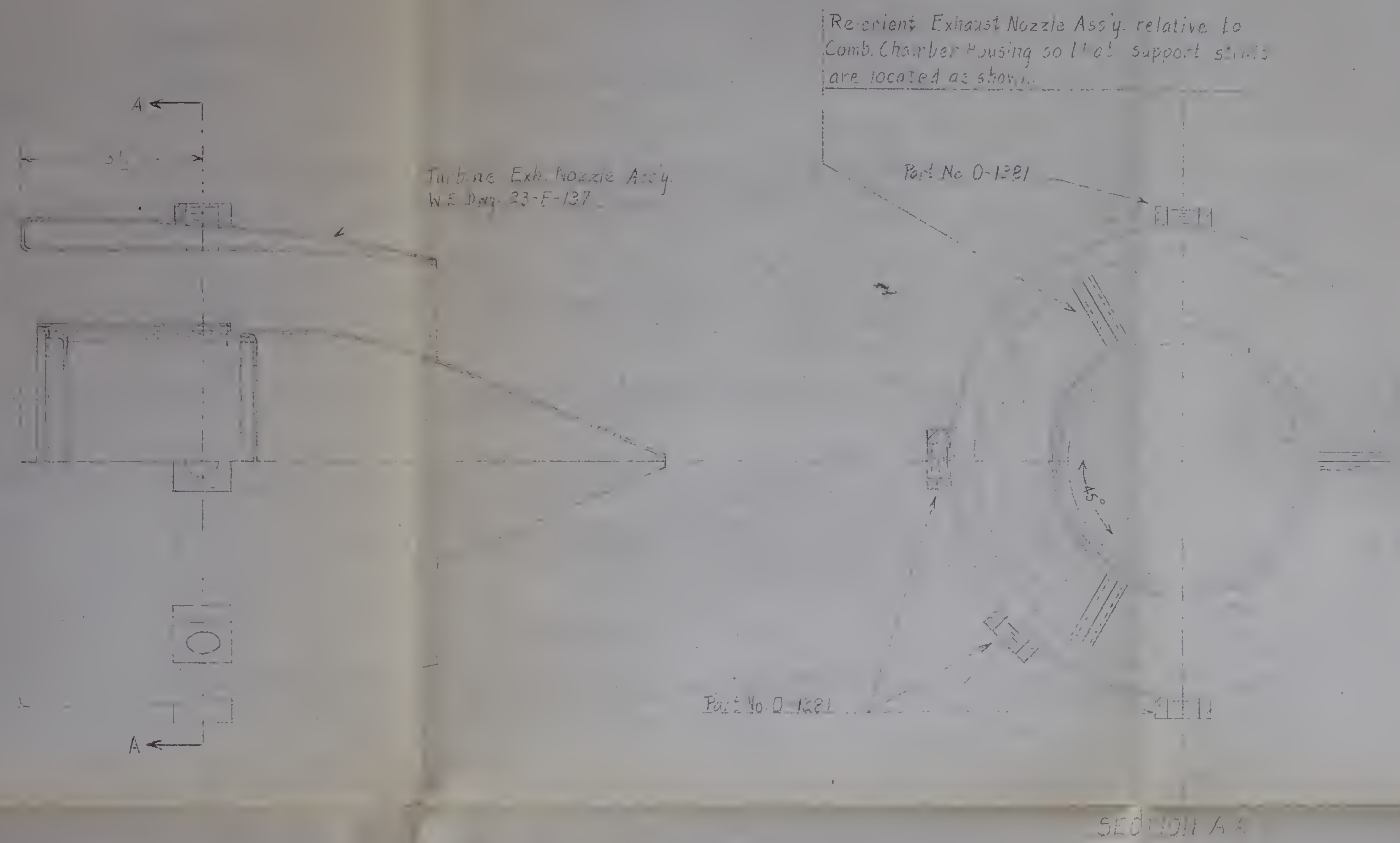
1	Mild Steel	1
NO.	MATERIAL	NO. REQ.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY

BEARING RETAINER RING

DRAWN BY [illegible]	DATE [illegible]
CHECKED BY	SCALE [illegible]
NEXT ASSEM [illegible]	[illegible]

FIG. 26



Showing Location for Instrument Access Pad (Part No 0-1281) on Turbine Exhaust Nozzle (W.E. Dwg 23-F-137)
Silver-Solder or Braze Pad Blocks into place before drilling for Inst. Probe Bushings (Part No. 0-1279). Drill through all members to air stream. After trial assembly trim Bushing end to coincide with outer boundary of air stream.

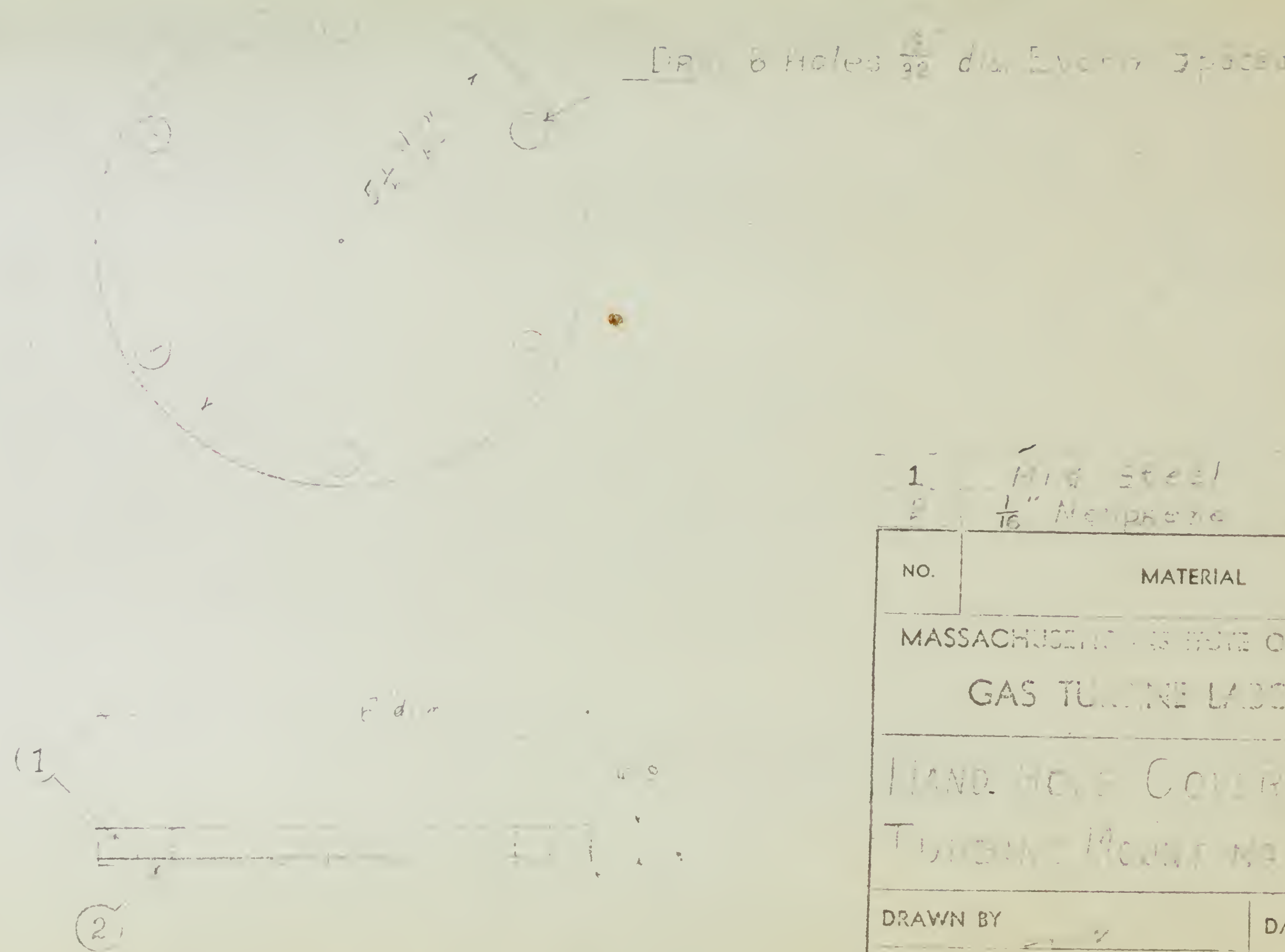
NO.	MATERIAL	REQ

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
GAS TURBINE LABORATORY

INSTRUMENT ACCESS LOCATIONS
TAIL - PIPE

DRAWN BY [illegible]	DATE 5/1-49
CHECKED BY	SCALE Full Size
NEXT ASSEM. C-5359	0-3283

FIG. 27



1	Mild steel	1
2	$\frac{1}{16}$" Neoprene	1
NO.	MATERIAL	NO REQ

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HAND HOLE COVER AND GASKET
[illegible]

DRAWN BY	DATE 5/[illegible]/49
CHECKED BY [illegible]	SCALE HALF [illegible]
NEXT ASSEM. C-4266	C-1286

FIG. 28